STEAMSHIP COEFFICIENTS, SPEEDS AND POWERS

STEAMSHIP COEFFICIENTS
SPEEDS AND POWERS

CONTAINING

THE DIMENSIONS AND PERFORMANCES OF VESSELS AND MANY PROGRESSIVE TRIALS OF SHIPS AND MODELS; WITH NOTES ON FROUDE'S LAW OF COMPARISON, SKIN FRICTION CORRECTION, ENGINE EFFICIENCY, THE ADMIRALTY "CONSTANT" SYSTEM OF NOTATION, TAYLOR'S METHODS, THE ADMIRALTY FORMULA, THRUST HORSE POWER, TABLES FOR WAKE VALUES, WAKE NOTATIONS REDUCED TO ONE BASIS, PROPELLER BLADE STRENGTH, PROPELLER EFFICIENCY CURVES, PITCH CORRECTION FACTORS, ETC. ETC.

FOR THE USE OF

ENGINEERS, SHIPBUILDERS, NAVAL ARCHITECTS, SUPERINTENDENTS, AND DRAUGHTSMEN

BY

CHARLES F. A. FYFE

Member of the Institution of Engineers and Shipbuilders in Scotland;
Member of the Institution of Naval Architects

SECOND EDITION

WITH 68 PLATES

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Mechanics Dept

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PREFACE TO THE SECOND EDITION.

THE best course open to the author in preparing a second edition was to rewrite the work completely.

In this edition he has almost discarded the method used in the first edition of reducing ships to 100-ft. models, and has adopted the more usual methods of presenting dimensions, proportions, and results of tests. Rear-Admiral Taylor and Professor Sadler give residuary resistance in lbs. per ton of displacement, while Mr Froude and Mr Baker employ the "Constant System of Notation" devised by Mr R. E. Froude, and used at the British Admiralty Experiment Tank Works and at the National Physical Laboratory Tank at Teddington.

Although the classic work of Mr R. E. Froude holds as a sound basis for the whole subject, yet until the results of tests of a sufficiently wide range of typical merchant-ship forms using his notation have been published, it has been found convenient at the present stage to use Taylor's residuary resistance per ton of displacement on a base of speed-length ratio, or E.H.P. upon a base of speed in knots, with tables for skin h.p.—the latter a necessary provision from the fact that merchant-ship forms often lie outside of the limits of Mr Taylor's curves for skin friction.

Recent model experiments have had a marked effect upon the design of ship forms, especially with regard to the longitudinal distribution of displacement.

The different notations for wake have been brought into line before tabulating values.

The importance of wind resistance has been emphasised, and an approximate method of calculation has been embodied.

The resistance of underwater appendages as a percentage of the total resistance has been included.

The author desires to acknowledge assistance kindly given by The Booth Steamship Company, Mr Geo. M. Welsh, Professor T. B. Abell, Mr A. T. Wall, and the help afforded by various books and other sources of information referred to in the text.

LIVERPOOL, *January* 1920.

PREFACE TO THE FIRST EDITION.

IN the following pages an attempt has been made to illustrate some of the practical uses of Froude's Law of Comparison. The data collected has been taken principally from the papers of some of the most eminent naval architects who have contributed towards the improvement of sound methods of comparing steamship performances. A few imaginary unnamed vessels have been included, derived in order to avoid using private trial data, and for the purpose of completing the lists of types. Where these occur, the word "actual" may be taken to mean "full sized."

The importance of considering the ratio of beam to length, in all questions of fineness appropriate to speed, has been emphasised throughout. The relations indicated in Plates 14 and 15 may be modified by a proper adjustment of this factor.

Plates 17 to 29 may be called "Rate Curves." The value of an ordinate of any of these curves, corrected for friction by Plates 8 to 11, and multiplied by $l^{3.5}$ from Table III, gives the power of any vessel for the corresponding speed. The result may be checked by the curves of Admiralty constant on Plates 4, 5, and 6, two-thirds powers of displacements up to 61 000 being given in Table IV.

Tables VII and VIII give skin friction horse-power and resistance for ships up to 700 ft. long at speeds up to 32 knots, and the explanatory matter generally deals with points omitted in other books.

Though aware of the shortcomings of the book, the author ventures to hope that it will be useful to practical men.

LIVERPOOL, 1907.

NOTATION FOR LAW OF COMPARISON.

	Full-sized ship.	Smaller ship or model.
Length, breadth, and mean draught in feet	L, B, H	l, b, h
Displacement in tons	D or Δ	d or δ
Speed in knots	V	v
Resistance following Froude's Law (*i.e.* residuary resistance)	R	r
Ratio of linear dimensions $\frac{L}{l} = \lambda$ . .	$L = \lambda l$	$b = \frac{B}{\lambda}$ $l = \frac{L}{\lambda}$
Ratio of speeds	$V = v\sqrt{\lambda}$ $\frac{V}{v} = \sqrt{\lambda}$ $V = v\left(\frac{\Delta}{\delta}\right)^{\frac{1}{6}}$	$v = \frac{V}{\sqrt{\lambda}}$ $v = \frac{V}{\left(\frac{\Delta}{\delta}\right)^{\frac{1}{6}}}$
Resistance per ton of displacement .	$\frac{R}{\Delta} = \frac{r}{\delta}$	$\frac{r}{\delta}$
Ratio of displacements	$\frac{\Delta}{\delta}\left(\frac{L}{l}\right)^3 = \lambda^3$	$\frac{\Delta}{\lambda^3}$
Ratio of residuary resistances . .	$R = \left(\frac{L}{l}\right)^3 r = \lambda^3 r$ $\lambda = \left(\frac{D}{d}\right)^{\frac{1}{3}}$	$r = \frac{R}{\lambda^3}$ $\frac{R}{r} = \frac{D}{d} = \frac{\Delta}{\delta}$
Piston speed of engine or peripheral speed turbines	$S = s\sqrt{\lambda}$	$s = \frac{S}{\sqrt{\lambda}}$
Revolutions per minute	$R = \frac{r}{\sqrt{\lambda}}$	$r = R\sqrt{\lambda}$
Piston areas	$A = a\lambda^2$	$a = \frac{A}{\lambda^2}$
Piston load	$W = w\lambda^3$	$w = \frac{W}{\lambda^3}$
Steam pressure	$P = \lambda p$	$p = \frac{P}{\lambda}$
Effective horse-power	$EHP = ehp\lambda^{3.5}$	ehp
Torque	$T = t\lambda^{3.5} \div \frac{1}{\lambda^{\frac{1}{2}}}$	t
Pressure of water in which propeller works	$P = \lambda p$	p
Thrust	$T = t\lambda^3$	
Residuary effective horse-power . .	$EHP_r = ehp_r\lambda^3$	ehp_r
Wetted surface $= C\sqrt{\Delta L}$ (where C is a coefficient based on shape, proportions, etc. See Taylor).	$WS = ws \times \lambda^2$	ws

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STEAMSHIP COEFFICIENTS, SPEEDS AND POWERS.

CHAPTER I.

INTRODUCTION.

THE LAW OF COMPARISON.

THE object of this publication is to provide shipbuilders and shipowners with a collection of actual results, and a proper method of comparing them, for reference when determining the power necessary to propel a proposed ship at a certain speed, and the fineness of form appropriate to that speed. The method of comparison is simply that of Froude, to whom the fundamental principles of the subject of marine propulsion are largely due. Instead of making an estimate of power founded upon calculation independent of experience, as is possible in mechanical engineering, practical estimators work from a store of data of previous steamship performances. The vessels selected for comparison with the proposed ship must be as far as possible "similar," having "similar speeds." By similar we mean that they have the same ratios of beam to length, and of draught to length, and the same coefficients of fineness. If we have two ships whose linear dimensions are similar, having equal block coefficients, their displacements are in the ratio of the cubes of their respective lengths. By "similar speeds" we mean speeds proportional to the square roots of the lengths of the vessels.

The data consist of progressive speed and power curves obtained from well-conducted progressive trials on the measured mile or in the open sea, at the normal draught and trim, or curves deduced from experimental tank trials of a model of the proposed vessel. From such curves the power at the "similar speed" may be obtained by inspection, and data in this form are

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much better than a collection of isolated results of trials at about full speed, with which, although each may be an accurate statement of the power at some stated speed, the speed or speed-length-ratio may not be the one we want. However, by the aid of some proper method of comparison, which will enable us to turn all results to useful account, we may make a fairly correct estimate of the power for our proposed steamer even from the latter.

The Law of Comparison has already been partly stated above. It is sometimes described as the Theory of Mechanical Similitude, or Froude's extended Law of Comparison.

The principle of similitude, first enunciated by Sir Isaac Newton, and proved last century by French mathematicians, M. Reech and others, will be found deduced mathematically by Rear-Admiral D. W. Taylor, U.S.N., in his *Speed and Power of Ships*, the book which contains the well-known and widely used curves of residuary resistance per ton of displacement.

The corresponding speeds for similar ships are speeds proportional to the square roots of their linear dimensions.

The corresponding displacements of similar ships are displacements proportional to the cubes of their linear dimensions.

The corresponding residuary resistances for similar ships at similar speeds are resistances proportional to the cubes of their linear dimensions.

The corresponding horse-powers required to overcome the residuary resistances for similar ships at similar speeds are powers proportional to the 3·5 powers of their linear dimensions.

The corresponding wetted surfaces and immersed midship areas of similar ships are proportional to the squares of their linear dimensions.

The Law of Comparison strictly applies to resistances other than frictional.

If the linear dimensions of an actual ship be l times the dimensions of a model (*i.e.* if the length of the ship be l times the length of the other ship or model), and the residuary resistances of the model at speeds V_1, V_2, V_3, etc., are R_1, R_2, R_3, etc., and the residuary horse-powers of the model at those speeds are h.p.$_1$, h.p.$_2$, h.p.$_3$, then at the corresponding speeds of the ship $V_1\sqrt{l}$, $V_2\sqrt{l}$, $V_3\sqrt{l}$, etc., the residuary resistances of the ship will be $R_1 l^3$, $R_2 l^3$, $R_3 l^3$, etc.; and the corresponding horse-powers to overcome the residuary resistance of the ship will be respectively h.p.$_1 l^{3\cdot5}$, h.p.$_2 l^{3\cdot5}$, etc.

A large part of the resistance of a ship or model in moving through the water, when either submerged or on the surface, consists of skin frictional resistance—about half to seven-eighths

of the total resistance according to whether the speed-length-ratio or speed is high or low, as will be seen by the analyses of trials later in the book. The remainder of the resistance is called the residuary resistance in the case of a model. Nearly all of the residuary resistance is wave-making resistance, but eddy-making very frequently accounts for about 8 per cent. of the total resistance of a ship or model. A full-sized ship encounters air or wind resistance, which, in the table on p. 5, we have included under the heading Residuary Resistance.

By the term "resistance" we mean the pull on the tow-rope registered by means of a properly arranged dynamometer, when towing a ship or model through still water. A sure method of determining the resistance of any ship is to tow her through still water from a long outrigged boom, at various speeds, and note the resistances.* The resistances of a ship towed at various speeds may also be inferred from trials of a small model of the ship in a tank in the light of Froude's method of proportioning the skin friction of the model to that of the ship.

Let R = the resistance of the ship in lbs. at any given speed.
E.H.P. = effective horse-power. This usually refers to the E.H.P. (naked), *i.e.* of the naked hull without bilge keels, bossings, and air resistance.
V = the speed of the ship in knots.

Then

$$R = \frac{\text{E.H.P.} \times 33\,000}{\text{Speed in feet per minute}}$$

$$R = \frac{\text{E.H.P.} \times 33\,000}{V \times 101{\cdot}33}$$

or

$$R = \frac{\text{E.H.P.} \times 60 \times 33\,000}{V \times 6080}$$

$$R = \frac{\text{E.H.P.} \times 325{\cdot}66}{V}$$

$$R = \frac{\text{E.H.P.}}{V \times {\cdot}003\,070\,7}.$$

E.H.P. is the equivalent of resistance. It is the horse-power expended in overcoming the net resistance of the vessel.

* See Mr A. T. Wall's paper, *Transactions Liverpool Engineering Society*, 1917, on "The Need for Research Work on the Propulsion of Ships."

$$\text{E.H.P.} = \frac{\text{Resistance in lb.} \times \text{speed of ship in feet per min.}}{33\,000}$$

or

$$\text{E.H.P.} = \frac{\text{Resistance in lb.} \times \text{speed in knots} \times 6\,080}{60 \times 33\,000}$$

or

$$\text{E.H.P.} = \text{Resistance in lb.} \times \text{speed in knots} \times 0{\cdot}003\,07.$$

Though towing trials and tank experiments of models of old steamers had no other use than to show these ratios of $\frac{\text{E.H.P.}}{\text{I.H.P.}}$, they would be valuable for enabling us to predict the speed of any given steamer, attainable by a given I.H.P. On Plates 1, 2, 3 will be found curves of this ratio $\frac{\text{E.H.P.}}{\text{I.H.P.}}$, or propulsive efficiency, or propulsive coefficient as it is sometimes called.

The propulsive efficiency is the product of three efficiencies, viz. (*a*) the engine efficiency, (*b*) the propeller efficiency, and (*c*) the hull efficiency. While the numerator is naked resistance, the denominator includes engine and propeller losses, and losses due to the interaction of hull and propeller.

An idea of engine efficiency can be got from ratios of Brake Horse-power to Indicated Horse-power in smaller engines, or from torsion meter measurements applied to turbine shafts, the ratio of the shaft horse-power to the I.H.P. being the engine efficiency. If we know the mechanical efficiency of a reciprocating engine at different speeds, we may deduct the power expended in overcoming the friction of the engines and line shafting from the gross I.H.P., and call the remainder the power delivered to the propeller, D.H.P. (Delivered Horse-power). See p. 143. T.H.P. = thrust horse-power of the screw.

The propeller efficiency for various slip ratios and pitch ratios is shown on Plates 55 to 63, plotted from the tables in Mr R. E. Froude's 1908 paper to the Institution of Naval Architects.

The D.H.P. divided by this figure gives the T.H.P. Working down, the T.H.P. ÷ hull efficiency = E.H.P. In design it is better to begin with E.H.P. and work up.

FACTORS CAUSING VARIATIONS IN THE TOTAL NET RESISTANCE OR TOW-ROPE RESISTANCE OF A SHIP.

SKIN FRICTIONAL RESISTANCE.

An estimate can always be made, based on the formula $f \times \cdot 80103 \times V^n$, where f depends upon length, quality of surface, density of fluid, and temperature, and n depends on speed and f.

- **HULL PROPER**, including an ordinary amount of deadwood. Effect of foul bottom. The condition of the immersed surface will determine the values of the constants to be used.
- **SURFACE OF APPENDAGES**, such as bilge keels, propeller struts, shaft bossings, and deadwood in excess of the usual amount.
- **MR BAKER'S PERCENTAGE* ADDITION** (5% to 20%), over the experimental results tabulated by Froude for planes, to allow for the increase in mean rubbing velocity between the streams and the ship, depending upon the fulness of the form.

$\left(f \times \text{wetted surface} \times \cdot 30707 \times V^n.\right) + \left(\text{A percentage addition for appendages.}\right) + \left(\text{A percentage addition to the previous total.*}\right)$

(Captain Dyson's book gives percentages.)

* See note on p. 6.

RESIDUARY RESISTANCE.

Cannot be calculated independently with exactness, but is obtained by subtracting the Skin Frictional Resistance from the Total Resistance.

- **EDDY-MAKING RESISTANCE**, due to irregular motion of rudder, water round propeller struts, broken water round the stern-post, stem, bilge keels and (a minor factor) other appendages.
- **WAVE-MAKING RESISTANCE.** By far the largest factor in the residuary resistance. Determined almost solely by (1) the shape of the curve of cross-section area, taken with the prismatic coefficient; (2) the extreme beam; (3) the surface water-line of the fore-body.
  Wave-making resistance is affected by (*a*) currents, banks, and shoals; (*b*) changes of trim (slight); (*c*) rolling and pitching, involving retardations; (*d*) rough water tending to disturb the regular formation of waves; (*e*) rolling and pitching placing the ship in positions which cause the total average resistance to be increased. Good effects of steadiness and large size. (Taylor's Contours of Residuary Resistance may be used.)
- **AIR RESISTANCE.** Affected by differences in the force of the wind, and differences in the amount of hull above water and exposed to wind. (Mr Taylor's formula $R_A = \cdot 0043\,AV^2$ may be used.)

We have

$$\rho = \frac{\text{E.H.P.}}{\text{I.H.P.}} = \underset{\substack{\text{(Engine}\\\text{efficiency)}}}{e_1} \times \underset{\substack{\text{(Screw}\\\text{efficiency)}}}{e_2} \times \underset{\substack{\text{(Hull}\\\text{efficiency)}}}{e_3}$$

(*a*) The *engine efficiency* may be taken at 0·83 at sea (*i.e.* at about seven-eighths full power), for engines driving their own air, circulating, feed, and bilge pumps, and about 0·84 to 0·845 at maximum power.

When only the air, feed and bilge pumps are driven from the main engine levers, we may take the engine efficiency at about 0·85 at sea, for good engines running at 600 to 700 ft. per minute piston speed at sea, and 0·86 at maximum trial power.

With all the pumps independent of the main engines the mechanical efficiency may be assumed 1 per cent. more, and with forced lubrication, as in some 1st class cruisers launched in 1906, e_1 would be say 0·88, or nearly 0·89. Professor Peabody puts down 0·886 for the "Manning" at full power. This is about a maximum. In large electric installations higher figures are given—over 0·91; but in marine work the values of e_1 given above may be adhered to, especially the 0·85 at sea for present-day reciprocating engines with centrifugal circulating pumps. For notes on Thrust Block Friction, see Appendix.

(*b*) *Propeller efficiency* (e_2) for various slip ratios and pitch ratios is shown on curves on Plates 55 to 63, plotted from the tables in Mr R. E. Froude's 1908 paper to the I.N.A.

(*c*) *Hull efficiency* $= \dfrac{\rho}{e_1 \times e_2} = e_3$. Only obtainable from tank trials.

A method of comparison in which the different ships or models were converted to a standard length of 100 ft. was suggested by

* From Mr Baker's book we note the following (see also p. 34) :—

	Fine battleships.	Merchant ships.		
		Fine.	Medium.	Full.
Prismatic coefficient . .	·60	·66	·75	·82
Mean excess of measured resistance over skin resistance, calculated from W. Froude's results	·10	·10	·135	·21

Mr F. P. Purvis in 1880 (*Trans. Inst. Engineers and Shipbuilders in Scotland*), and elaborately worked out by Mr W. Hök in an admirable paper read before the North-East Coast Institution of Engineers and Shipbuilders in 1893, was adopted in our first edition. Mr Hök's results were all expressed in progressive speed curves with i.h.p. per ton of displacement of the 100-ft. model, the i.h.p. being corrected for engine friction; so that the power at lower speeds of any ship may be considered that of an engine designed for that power working at maximum efficiency, instead of being, like ours, ordinary progressive speed and i.h.p. curves taken directly from trials, and suffering slightly at low speeds from the decreased engine efficiency. Mr Hök's curves of i.h.p. and speed are therefore steeper than curves of quantitative results reduced by the Law of Comparison.

By the Theory of Mechanical Similitude the relation between the resistance and speed of a ship can be found from the trial results of a "similar" ship as follows:—

A ship 315 ft. long × 40 ft. broad × 15·8 ft. mean draft, of 3 400 tons displacement, is "similar" to a ship 325 ft. long × 41·3 ft. broad × 16·3 ft. mean draft, of 3 740 tons displacement. If the residuary resistance of the smaller ship at 18 knots speed is 30 500, then at 18·29 knots, the "corresponding" speed of the larger vessel, the residuary resistance for the 325-ft. ship will be greater than 30 500 in the proportion of the cubes of the lengths of the two ships.

That is, the residuary resistance of the larger vessel will be

$$30\,500 \times \frac{34 \cdot 33}{31 \cdot 25} = 33\,500 \text{ lbs.}$$

By the Law of Comparison the corresponding horse-powers required for overcoming the residuary resistances are proportional to l to the power 3·5. Thus in this example the residuary horse-powers will be 1 685 and 1 881, in the proportion of 55·4 to 61·9.

The skin resistance is calculated separately.

Any number of ships may be derived from these figures all exactly similar, differing from each other only in mere size.

In the method of 100-ft. models the principal characteristics of their immersed forms are displayed with more readiness than by perhaps any other method. The breadth and draught of the ship are then percentages of the length. Thus in the above cases, the two 100-ft. models are 100 × 12·7 × 5·02, with a displacement of 109 tons, and a speed of 10·15 knots in both cases, and the block coefficient is 0·598.

Let the ratio of the length of the actual ship to the length of the reduced ship be l; then in this book

$$l = \frac{\text{Length of ship}}{100}.$$

In manipulating the data in connection with the above ships, $l = 3{\cdot}15$ and $3{\cdot}25$ for the two cases.

A table of square roots, squares, cubes, and $3\frac{1}{2}$ powers of values of l from 0·25 up to say 8·00, will help us to handle such data with great ease. (See Table XIII, pp. 38–46.) This table serves the later methods of comparison, though it was originally prepared for the 100-ft. models.

The wetted surface of the 100-ft. model (by Mumford's formula) $= (100 \times 5{\cdot}02 \times 1{\cdot}7) + (100 \times 12{\cdot}7 \times 0{\cdot}598) = 1613$ square ft.

Turning to the curves of skin horse-power correction (Plates 3 to 6), we find that 315-ft. ships reduced to 100-ft. models, at 10·15 knots require 3 horse-power per 1 000 square ft. of wetted surface of a correction for their skin horse-power.

Therefore we make a correction of $1{\cdot}613 \times 3 = 4{\cdot}839$ H.P.

Application of the Law of Comparison.

Given the particulars of a destroyer:—Length, 212 ft. Beam, 19·75 ft. Mean draught, 6·5 ft. Block coefficient = ·386. Wetted surface = 3 970 sq. ft. Displacement = 300 tons. Total resistance at 15·8 knots speed = 3·5 tons. $\frac{V}{\sqrt{L}} = 1{\cdot}085$.

From this let us deduce the speed and power of a cruiser of similar form, 765 ft. long.

Call the ratio of the length l, then $l = \frac{765}{212} = 3{\cdot}61$.

The ratio of the displacements $= l^3 = 47{\cdot}04$.

The ratio of the corresponding speeds $= \sqrt{l} = 1{\cdot}9$.

The ratio of the wave-making resistances at these speeds $= l^3 = 47\ 04$.

The ratio of the wetted surfaces $= l^2 = 13{\cdot}03$.

The ratio of the residuary horse-powers $= l^{3\cdot5} = 89{\cdot}5$.

From this we find that the cruiser is $765 \times 71{\cdot}3 \times 23{\cdot}5$ ft. mean draught. Wetted surface = 51 600 sq. ft. Displacement = 14 100 tons. Speed = 30 knots.

The total E.H.P. of the destroyer

$$\begin{aligned}
&= \text{lbs. resistance} \times \text{speed} \times \cdot 003\,07. \\
&= (3\cdot 5 \times 2\,240) \times 15\cdot 8 \times \cdot 003\,07. \\
&= 381.
\end{aligned}$$

Skin H.P. of destroyer $= 3\cdot 970 \times 70 = 278$.
(Using Table IX, p. 31.)

$\therefore$ Residuary H.P. of destroyer $= 103$.

Residuary resistance of destroyer $= \dfrac{103}{15\cdot 8 \times \cdot 003\,07} = 2\,122$ lbs.

Residuary resistance of cruiser $= 212\,2 \times 47\cdot 04 = 100\,000$ lbs.

Residuary H.P. of cruiser

$$\begin{aligned}
&= \text{lbs. residuary resistance} \times \text{speed} \times \cdot 003\,07. \\
&= (100\,000) \times 30 \times \cdot 003\,07. \\
&= 9\,210.
\end{aligned}$$

Skin H.P. of cruiser $= 51\cdot 6 \times 415 = 21\,420$.
(Using Table IX, p. 32.)

Total effective H.P. of cruiser at 30 knots

$$\begin{aligned}
&= \text{skin H.P.} + \text{residuary H.P.} \\
&= 21\,420 + 9\,210. \\
&= 30\,630.
\end{aligned}$$

A quicker way to arrive at the residuary H.P. of the cruiser is to simply multiply the residuary H.P. of the destroyer by $89\cdot 5$.

Calculation of wetted surface of destroyer by Froude's formula :—

$$S = (\Delta 35)^{\frac{2}{3}}\left(3\cdot 4 + \frac{L}{2(\Delta 35)^{\frac{1}{3}}}\right)$$

$$S = (35 \times 300)^{\frac{2}{3}}\left(3\cdot 4 + \frac{212}{2 \times (35 \times 300)^{\frac{1}{3}}}\right)$$

$$= (10\,500)^{\frac{2}{3}}\left(3\cdot 4 + \frac{212}{2 \times (10\,500)^{\frac{1}{3}}}\right)$$

$$= 479\cdot 49\left(3\cdot 4 + \frac{212}{2 \times 22}\right)$$

$$= 479\cdot 49\left(3\cdot 4 + \frac{212}{44}\right)$$

$$= 479\cdot 49 \times (3\cdot 4 + 4\cdot 84)$$

$$= 479\cdot 49 \times 8\cdot 24$$

$$= 395\,0,$$

D = displacement of ship in tons.
D_m = " 100-ft. model in tons.
V = speed of ship in knots.
V_m = corresponding speed of 100-ft. model in knots.
I.H.P., E.H.P., T.H.P. = indicated horse-power, effective horse-power, and thrust horse-power respectively, for full-sized ship.
i.h.p., e.h.p., and t.h.p. = ditto for 100-ft. model.
Revs. = Revolutions per min. in the case of actual ship.
Revs._m = " " " 100-ft. model.
$(\text{Indicated thrust})_m$ = Indicated thrust for 100-ft model.
$(\text{Resistance})_m$ = Resistance of 100-ft. model.
L = length of ship in feet.
l = the ratio of the length of the ship to the length of the reduced ship or 100-ft. model; i.e. $l = \frac{100}{L}$.
ω = block coefficient (same for both).

The relations are expressed by the following formulæ :—

$$D_m = \frac{D}{l^3}.$$

$$V_m = \frac{V}{\sqrt{l}}\text{; Revs.}_m = \text{Revs.}\sqrt{l}\text{; e.h.p.} = \frac{\text{E.H.P.}}{l^{3\cdot5}},$$

with skin friction correction where necessary.

$$(\text{Wetted surface})_m = \frac{\text{Wetted surface of actual ship}}{l^2}.$$

Resistance and thrust vary as l^3 for corresponding speeds.

Horse-power varies as $l^{3\cdot5}$ for corresponding speeds, with skin friction correction where necessary.

E.H.P. = resistance in lb. × V × 0·003 070 7.

Skin resistance = R_f = coef. of friction × wetted surface × V^n.

Skin horse-power = f × wetted surface × 0·003 070 7 × $V^{2\cdot83}$.

Skin horse-power = skin resistance × (0·003 070 7 × V).

For values of f (the coefficient of friction) and the index n, see Tables I–VII, pp. 17–26, and Plate 7.

For a ship 500 ft. long, f = 0·009 04, and n = 1·83.

The coefficients of fineness are the same for both ship and model.

It may be noted that

$$\text{Mid-area coefficient} = \frac{\text{Block coefficient}}{\text{Prismatic coefficient}}.$$

$$\text{Prismatic coefficient} = \frac{\text{Block coefficient}}{\text{Mid-area coefficient}}.$$

Mumford's formula for Wetted Surface, given by Sir A. Denny (*Trans. Inst. Naval Arch.*, 1895), and used throughout this work.

Wetted Surface in square feet

$$= (\mathrm{L} \times \mathrm{D} \times 1{\cdot}7) + (\mathrm{L} \times \mathrm{B} \times \text{block coefficient}),$$

where L = length, B = breadth, D = draught, of ship in feet.

The surfaces obtained by this formula are almost exactly correct for steamers of medium fineness whose draught (100-ft. model) is 3·72 to 5·45, and beam from 10 to 14·44. Mid-area coefficient 0·913 to 0·94, and block coefficient 0·614 to 0·659. The percentage error for 28 steamers taken was not over $1\frac{1}{2}$ per cent. up or down.

For finer steamers the formula slightly overestimates, and for full steamers the reverse. With a very broad, full and shallow barge the formula gave Wetted Surface 3·36 per cent. too low.

For full steamers 1·8 or 1·9 may be required instead of 1·7.

Other formulæ for Wetted Surface are noted below.

Note on Humps.—The deeper the draught the higher are the speeds at which humps and hollows occur. Mr R. E. Froude, in his 1881 paper, gave the hump speeds and hollow speeds for a series of ships, see p. 118.

WETTED SURFACE.

(1) **Mumford's formula**, given by Sir A. Denny, in a paper to the Institution of Naval Architects, is reliable:—

$$\mathrm{L}(1{\cdot}7\mathrm{D} + \beta\mathrm{B})$$

or written thus,

$$(\mathrm{L} \times \mathrm{D} \times 1{\cdot}7) + (\mathrm{L} + \beta \times \mathrm{B}),$$

where L = length of ship in feet between the perpendiculars, or mean immersed length in the case of cruiser sterns.
D = draught of ship in feet.
B = breadth " "
β = block coefficient.

For block coefficients over ·78, 1·7 may be altered to 1·8, and for extreme forms such as shallow-draft vessels 1·9 or 2·0 may be required.

(2) **Mr Froude's formula**, applicable to Admiralty types, is

$$\mathrm{S} = (\Delta 35)^{\frac{2}{3}}\left(3{\cdot}4 + \frac{\mathrm{L}}{2(\Delta 35)^{\frac{1}{3}}}\right),$$

where $(\Delta 35)$ = displacement in cubic feet.

(3) **Taylor's formula** :—

$$= S = C\sqrt{\Delta L},$$

where C = a coefficient from Taylor's curves, depending upon beam-draught ratio and midship section coefficient.
Δ = displacement in tons.
L = length in feet.

Example.—30-knot destroyer, 212 × 19·75 × 6·5 ft. mean draft. 300 tons displacement. Block coefficient = ·386 = w.

(1) **Mumford's formula** :—

$$\begin{aligned} & (L \times B \times w) + (L \times D \times 1{\cdot}7) \\ = & (212 \times 19{\cdot}75 \times {\cdot}386) + (212 \times 6{\cdot}5 \times 1{\cdot}7) \\ = & 3\,970 \text{ square ft.} \end{aligned}$$

(2) **Admiralty formula** :—

$$\begin{aligned} S &= (35 \times \Delta)^{\frac{2}{3}} \times \left(3{\cdot}4 + \frac{L}{2 \times (35 \times \Delta)^{\frac{1}{3}}}\right) \\ &= (10\,500)^{\frac{2}{3}} \times \left(3{\cdot}4 + \frac{212}{2 \times (10\,500)^{\frac{1}{3}}}\right) \\ &= 479{\cdot}49 \times \left(3{\cdot}4 + \frac{212}{2 \times 21{\cdot}9}\right) \\ &= 3\,950 \text{ square ft.} \end{aligned}$$

BLOCK COEFFICIENT.

"The ratio of the immersed volume of displacement of a vessel to the volume of the circumscribing parallelepipedon."

In 1911–12 the Institution of Engineers and Shipbuilders in Scotland, acting upon a resolution passed during the discussion of a paper read in 1910 by Mr P. A. Hillhouse, entitled "The Block Coefficient," appointed a committee to frame definitions of coefficients of displacement.

In the report the committee recommended the name "Coefficient of Fineness," giving it the standard symbol C.F.

Their recommendations were as follows :—

$$\text{C.F.} = \frac{\times\ 35}{L \times B \times D}.$$

Δ = Displacement at load draught, inclusive of shell-plating, bosses, etc., as usually given on ship's displacement scale.
L = Length of vessel on load-line, *i.e.* the length from after side of stern-post to fore side of stern.
B = Moulded breadth plus the mean thickness of shell-plating on sides, *i.e.* three thicknesses of plate with out-and-in strakes and two thicknesses with joggled plating.
D = Moulded draught plus the mean thickness of shell-plating on bottom, *i.e.* $1\frac{1}{2}$ thicknesses of plate with out-and-in strakes and one thickness with joggled plating.

The report mentions that "the above formula complies with the conditions of the definition, and, in utilising easily obtained particulars, avoids discussion or calculation relative to allowance for appendages. No attempt is made to deal in detail with abnormal cases, such as vessels having sides out of the vertical, corrugated, or sponsoned, but it is considered that, by adhering to the spirit of the definition, and choosing the enclosing rectilinear figure, so that it holds a similar relationship to the enclosed form as the parallelepipedon bears to an ordinary vessel, there will be no practical difficulty in dealing with such cases. Such necessary departures from the basis formula should always be expressed by those who use and specify Coefficients of Fineness."

With a cruiser stern, L.W.L. may perhaps define the length better than length b.p.

Draught.

In Professor Durand's and Dr A. C. Kirk's data gross draught is quoted, *i.e.* a figure which includes the hanging keel. The coefficients, block and mid area, show what the net draught is. For instance, for "Bayern," 24·1 with keel is the draught given. The block coefficient and midship section coefficient show that the net draught of the hull form is 23·3 ft. In quoting a figure for prismatic coefficient, it is necessary to state whether the draught includes the hanging keel (if any) or whether the draught is taken to the bottom of the shell-plating.

Conversion Factors.

1 cubic metre = 35·316 6 cubic ft.
1 cubic foot = ·028 31 cubic metre.
1 ton (English or U.S. standard) = 2 240 lbs.
1 cwt. (English or U.S. standard) = 50·802 4 kilograms.

1 lb. (English or U.S. standard) = ·453 6 kilogram.
1 kilogram = 2·204 6 lbs.
1 gram = 15·432 356 4 grains.
*1 tonne or tonneau or millier (French) $\begin{cases} = 1\,000 \text{ kilograms.} \\ = 2\,204{\cdot}6 \text{ English lbs.} \end{cases}$
1 English gallon = ·160 4 cubic ft. = 4·545 963 1 litres.
1 metre (0° C.) = 39·370 113 inches (62° F.) = 3·280 ft.
1 square metre = 10·763 9 square ft.
1 litre = 1·759 8 pint.
1 cubic decimetre = 61·024 cubic inches.
1 yard = 0·914 399 metre.
1 cubic inch = 16·387 cubic centimetres.
1 pound (avoirdupois) = 0·453 592 43 kilogram.
1 kilogram per square centimetre = 14·223 2 lbs. per sq. in.
1 inch = 25·399 5 millimetres.
1 inch = 2·539 95 centimetres.
1 kilogram per square millimetre = 1 422·32 lbs. per sq. in.
1 kilogram per square centimetre = 14·222 lbs. per sq. in.
1 square inch = 6·451 336 square centimetres.
1 square inch = 645·137 square millimetres.
1 square foot = 928·997 square centimetres.
1 foot = 30·479 7 centimetres.
1 foot = 304·797 millimetres.
1 square foot = ·092 9 square metre.
1 square foot = 92 899·7 square millimetres.
1 cubic centimetre = ·061 027 cubic inch.
1 millimetre = ·039 4 inch.
1 centimetre = ·394 inch.

Number of kilograms per square millimetre × ·635 gives number of tons per square inch.

Number of tons per square inch × 1·575 = number of kilograms per square millimetre.

* In calculating displacement in metric tons (French), take 34·4 cubic feet salt water per ton.

CHAPTER II.

SKIN FRICTIONAL RESISTANCE.

WATER is not a perfectly frictionless liquid, but is viscous to a certain extent, and the wetted surface of a plank or ship, moving through water, carries a layer of water with it. In Professor Hele-Shaw's paper to the Institution of Naval Architects, 1898, the stream lines in the frictional belt of viscous fluid were plotted, and seemed to have a straight-line flow in contact with the moving body and a whirling motion at the outer boundary—the forces causing these motions being due to inertia. Professor Lamb's investigations, published in the same paper, indicated viscous resistance as the operating cause. At any rate, energy is imparted to the water, and this causes resistance to the motion of the vessel. The after end of the moving body rubs against water which has already been set in motion by the forward end, and therefore does not cause so much resistance from it, though the velocity of the layer of water is greater at the stern than at the bow. With a long plank probably the resistance at the rear end is only half that of its fore end. Mr G. S. Baker, in his Newcastle paper, 1915, on "Notes on Model Experiments," discussed the effect, on the resistance of the whole plank, of the forward momentum of the (wake) water at the rear end of the plank. There is very little whirling or vortex motion except in a rough plank, say, covered with barnacles. The average forward velocity of the frictional belt increases with the length of the immersed body, while the mean resistance per square foot decreases with increased length. In the British Association reports for 1872–74, Mr Wm. Froude gave curves showing, for flat wood planes in fresh water, the relation between length and resistance, enunciating the formula

$$R = fSV^n,$$

which approximately expresses the resistance at speed V of S square feet of surface, where f is the coefficient of friction de-

pending upon (1) the quality of the surface of the board; (2) the length of the surface (and it decreases at a decreasing rate as the length of the surface is increased); (3) the temperature, being about 3 or 4 per cent. less in summer than in winter; (4) the density of the fluid, due to the fact that skin resistance is really an eddy resistance.* The resistance in salt water is thus about $\frac{36}{35}$ times the resistance in fresh water. The value of n is not altogether independent of f, but, generally speaking, it depends upon the nature of the surface and usually decreases with length, and has different values for different speeds (Plates 1 and 2). A dirty surface, such as a weed-coated, barnacled, or shell-encrusted ship's bottom, may have its skin frictional resistance increased two-, three-, or even five-fold. Rear-Admiral Taylor states that a marine growth, consisting mostly of barnacles, averaging in total weight when dry only $\frac{1}{4}$ lb. per square ft., would increase the frictional resistance by 210 per cent.

In a paper by Naval-Constructor M'Entee on "The Variation of Frictional Resistance of Ships with Condition of Wetted Surface," mention is made of 300 per cent. increased skin resistance from effects of fouling. In a lecture at Newcastle in 1915, Mr G. S. Baker gave an account of experiments to show the effect of the edges and butts of shell-plating on the resistance. The importance of having flush-plating at the forward part of a ship was clearly demonstrated in a valuable appendix. See Mr A. W. John's remarks in the discussion.

W. Froude's values of f found from experiments with boards or planes, bare and also coated with various compositions, agree with those for paraffin. Herr B. Tideman's experimental results for planes or planks in fresh water are very similar to those of W. Froude, but his values of f and n extrapolated for longer surfaces in salt water give higher results by 4 or 5 per cent. than Froude's. Mr Baker mentions in his book that a clean ship's bottom, painted with anti-fouling composition, gives practically the same result in the model as a paraffin surface, while a surface of the roughness of calico gives nearly double the resistance. The results from the earlier experiments are indicated on Plate 7. The figure for a little weed or barnacle is shown on Table VI. A ship's bottom covered with shells has a still higher coefficient of friction. At the United States Experimental Model Basin at

* The law was deduced from the resistances of flat boards 19 inches in depth, varying from 1 ft. to 50 ft. in length. Mr W. Froude, Mr R. E. Froude, and others extended the curve to give surface frictional results for long ships (Tables. V–VII), and these figures are almost universally accepted. Large-scale experiments are, however, required to verify the quantities. Moderate corrosion and rough paint would produce high resistances.

Washington, the values employed for 20-ft. models of smooth wood are

$$f = \cdot 009\,67, \quad n = 1\cdot 854.$$

These constants will be found to give the results tabulated in column 3, table ix. of Taylor's *Speed and Power of Ships*, and are proportional to those which were used in preparing Mr Taylor's figs. 81 to 120. Mr R. E. Froude states that for the paraffin models used in his experiments about 1886, both the coefficient f and the exponent n are substantially the same as for a smooth-painted or varnished surface.

The table on p. 21 shows the skin frictional resistance of "Yorktown," 20-ft. model, calculated from various sets of constants.

TABLE I.—SURFACE FRICTION OF PARAFFIN MODELS IN FRESH WATER.

The index n taken as 1·94 throughout.

Length of model in feet.	Coefficient of friction f.	Length of model in feet.	Coefficient of friction f.
2	·011 76	12	·009 08
3	·011 23	12·5	·009 01
4	·010 83	13	·008 95
5	·010 50	13·5	·008 89
6	·010 22	14	·008 83
7	·009 97	14·5	·008 78
8	·009 73	15	·008 73
9	·009 53	16	·008 64
10	·009 37	17	·008 55
10·5	·009 28	18	·008 47
11	·009 20	19	·008 40
11·5	·009 14	20	·008 34

NOTE.—Tank models are usually from 8 to 20 ft. in length, and the coefficient of friction diminishes with the length of the surface, as above.

For varnish, smooth paint, or compositions, tinfoil, calico, and medium sand, take f and n from Plate 1.

TABLE II.—SKIN FRICTIONAL RESISTANCE CONSTANTS FOR PARAFFIN MODELS IN FRESH WATER.

Value of f from Froude's tables.

Speed.		1·94 power of speed in knots.	Skin resistance in lbs. per 10 sq. ft. of wetted surface for models of various lengths.		
Feet per min.	Knots.		11·951 ft. long. f = ·009 08.	12 ft. long. f = ·009 07.	20 ft. long. f = ·00834.
240	2·37	5·3	·481	·480 5	·442
300	2·962	8·23	·748	·746	·686
340	3·357	10·41	·946	·945	
360	3·558	11·63	1·057	1·054	
380	3·75	12·95	1·178	1·174	1·08
400	3·95	14·3	1·298	1·296	1·192
420	4·147	15·77	1·433	1·43	1·314
440	4·346	17·2	1·563	1·56	1·433
480	4·74	20·5	1·862	1·86	1·71
500	4·933	22·14	2·012	2·01	1·843
540	5·33	25·67	2·331	2·33	2·14
580	5·73	29·6	2·69	2·687	2·47
600	5·92	31·4	2·851	2·85	2·62
640	6·32	35·72	3·25	3·24	2·972
680	6·71	41·04	3·73		
720	7·11	44·8	4·165	...	3·735
760	7·51	49·65	4·51	4·51	4·14
800	7·9	54·85	4·98	4·97	4·57
850	8·396	61·7	5·61	5·605	5·15
900	8·89	69·1	6·28	6·27	5·76
960	9·48	78·2	7·1	7·1	6·52

TABLE III.—FOR SURFACE FRICTION OF MODELS IN FRESH WATER.

No.	1·854 power.	2·854 power.	1·94 power.	2·94 power.	No.	1·854 power.	2·854 power.	1·94 power.	2·94 power.
1	1·0	1·0	1·0	1·0	5·5	23·58	129·7	27·29	150·2
1·2	1·40	1·68	1·42	1·71	5·6	24·35		28·26	
1·4	1·87	2·61	1·92	2·69	5·7	25·16		29·25	
1·5	2·12	3·18	2·20	3·30	5·8	26·02		30·26	
1·6	2·39	3·82	2·49	3·98	5·9	26·86		31·29	
1·8	2·98	5·36	3·13	5·64	6·0	27·71	166·3	32·33	194·0
2·0	3·61	7·23	3·84	7·67	6·1	28·57		33·39	
2·1	3·96	8·32	4·22	8·86	6·2	29·46		34·46	
2·2	4·32	9·51	4·62	10·20	6·3	30·35		35·54	
2·3	4·68	10·77	5·03	11·57	6·4	31·26		36·64	
2·4	5·07	12·16	5·46	13·11	6·5	32·15	209·0	37·76	245·5
2·5	5·47	13·67	5·92	14·79	6·6	33·07		38·89	
2·6	5·88	15·29	6·38	16·60	6·7	34·00		40·04	
2·7	6·31	17·03	6·86	18·55	6·8	34·95		41·21	
2·8	6·75	18·89	7·37	20·64	6·9	35·91		42·40	
2·9	7·20	20·88	7·89	22·88	7·0	36·88	258·2	43·60	305·2
3·0	7·66	23·00	8·43	25·28	7·1	37·87		44·82	
3·1	8·15	25·35	8·98	27·84	7·2	38·86		46·05	
3·2	8·64	27·65	9·55	30·56	7·3	39·87		47·30	
3·3	9·15	30·19	10·14	33·45	7·4	40·88		48·56	
3·4	9·67	32·87	10·74	36·52	7·5	41·91	314·4	49·84	373·8
3·5	10·20	35·71	11·36	39·77	7·6	42·98		51·14	
3·6	10·75	38·70	12·00	43·20	7·7	44·02		52·45	
3·7	11·31	41·84	12·66	46·83	7·8	45·08		53·78	
3·8	11·88	45·15	13·33	50·65	7·9	46·15		55·13	
3·9	12·47	48·63	14·02	54·67	8·0	47·24	377·9	56·49	451·9
4·0	13·07	52·27	14·72	58·88	8·1	48·34		57·87	
4·1	13·68	56·09	15·45	63·32	8·2	49·45		59·27	
4·2	14·29	60·08	16·18	67·98	8·3	50·57		60·68	
4·3	14·94	64·23	16·94	72·85	8·4	51·71		62·10	
4·4	15·59	68·61	17·71	77·94	8·5	52·86	449·3	63·54	540·1
4·5	16·26	73·16	18·50	83·26	8·6	54·02		65·00	
4·6	16·93	77·90	19·31	88·82	8·7	55·19		66·47	
4·7	17·62	82·83	20·13	94·62	8·8	56·37		67·96	
4·8	18·33		20·95		8·9	57·56		69·47	
4·9	19·03		21·82		9·0	58·77	528·9	71·00	639·0
5·0	19·76	98·82	22·70	113·5	9·1	59·99		72·53	
5·1	20·48		23·59		9·2	61·22		73·08	
5·2	21·24		24·50		9·3	62·46		74·65	
5·3	22·00		25·41		9·4	63·71		76·24	
5·4	22·77		26·34		9·5	64·97	617·2	78·85	749·0

TABLE III—*continued.*

No.	1·854 power.	2·854 power.	1·94 power.	2·94 power.	No.	1·854 power.	2·854 power.	1·94 power.	2·94 power.
9·6	66·24		80·46		11·6	94·08		116·1	
9·7	67·52		82·09		11·7	95·59		118·1	
9·8	68·82		83·74		11·8	97·11		120·1	
9·9	70·13		85·41		11·9	98·65		122·1	
10·0	71·45	714·5	87·10	871·0	12·0	100·2	1 202·2	124·1	1 488·7
10·1	72·78		88·79		12·1	101·8		126·1	
10·2	74·12		90·50		12·2	103·3		128·1	
10·3	75·47		92·23		12·3	104·9		130·1	
10·4	76·83		93·98		12·4	106·5		132·2	
10·5	78·21	821·2	95·74	1 005·3	12·5	108·1	1 350·8	134·3	1 678·5
10·6	79·60		97·52		12·6	109·7		136·4	
10·7	81·00		99·31		12·7	111·3		138·5	
10·8	82·41		101·1		12·8	112·9		140·6	
10·9	83·83		102·9		12·9	114·6		142·7	
11·0	85·26	937·9	104·8	1 152·6	13·0	116·2	1 510·8	144·9	1 883·6
11·1	86·70		106·6		13·1	117·9		147·1	
11·2	88·15		108·5		13·2	119·5		149·25	
11·3	89·61		110·4		13·3	121·2		151·5	
11·4	91·09		112·3		13·4	122·9		153·7	
11·5	92·58	1 064·7	114·2	1 313·6	13·5	124·6	1 682·6	155·9	2 104·7

TABLE IV.—FOR SURFACE FRICTION OF MODELS IN FRESH WATER.

Length of model in feet.	Coefficient of friction		Length of model in feet.	Coefficient of friction	
	f.	*n.*		*f.*	*n.*
8	·010 55	1·854	20	·009 67	1·854
9	·010 45	,,	21	·009 64	,,
10	·010 35	,,	22	·009 60	,,
11	·010 25	,,	23	·009 55	,,
12	·010 17	,,	24	·009 50	,,
13	·010 10	,,	25	·009 45	,,
14	·010 03	,,	26	·009 40	,,
15	·009 96	,,	27	·009 35	,,
16	009 90	,,	28	·009 32	,,
17	·009 84	,,	29	·009 30	,,
18	·009 79	,,	30	·009 28	,,
19	009 73	,,			

20-FT. MODEL OF "YORKTOWN."

Wetted surface = 72·46 sq. ft. δ = displacement in lbs. in fresh water = 2 405. v = speed in hundreds of feet per minute. R. E. Froude's (s) = 6·35, and O = ·114 70.

v = speed of 20-ft. model in knots.	Skin Frictional Resistance in lbs. Values from Taylor's table ix.	Values from f = ·009 67, n = 1·854.	Values from f = ·009 7, n = 1·854.	Values from Tideman's, f = ·008 34, n = 1·94.	Values from W. Froude's, f = ·008 8, n = 1·94.	Values from Tideman's salt water, f = ·010 57, n = 1·848 4.	Tideman's salt water, f = ·010 57, and Taylor's n = 1·83.	Values from Froude's constants for salt water, f = ·010 55, n = 1·825.	Values from R. E. Froude's, $OSL^{-.175}$.	v.	(C)
3	5·36	5·36	5·38	5·1	5·39	5·83	...	5·68	...		
4	9·15	9·16	9·2	8·9	9·4	9·94	9·76	9·55	9·35	4·0533	1·017
5	13·81	13·81	13·87	13·7	14·32	...	14·52	14·41	14·01	5·0666	1·311
6	19·39	19·44	19·5	19·5	20·58	...	20·3	19·9	19·54	6·0799	2·435
6·3	21·2	21·25	...	21·45	22·62						
	Taylor	A		B					C		

Taylor's results are calculated from the constants in column A.

The discrepancy in column C is probably chiefly due to the fact that Mr Taylor's total resistance, from which (C) is calculated, is given in round numbers.

It is important that correct values should be obtained by model experimenters for the skin resistance of the model, in order that correct residuary resistances may be obtained for application by the Law of Comparison to the full-sized ship. The smaller values of the skin friction of models are on the safe side, because they do not involve an underestimate of the residuary resistance.

"Yorktown," 20-ft. model (naked), on even keel. 230 × 36 × 13·82 ft. mean draft :—

$\frac{B}{H} = \frac{36}{13\cdot82} = 2\cdot61$. $\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 12\cdot167$. Midship section coefficient = ·868. Prismatic coefficient = ·592. Δ = 1 680 tons salt water. Wetted surface = 9 582 sq. ft.

230 is the "mean immersed length" of ship.

$$\frac{\text{Length of ship}}{\text{Length of model}} = \frac{L}{l} = \frac{230}{20} = 11\cdot5.$$

$$\frac{\text{Speed of ship}}{\text{Speed of model}} = \sqrt{\frac{L}{l}} = 3\cdot391.$$

$$\frac{\text{Wetted surface of ship}}{\text{Wetted surface of model}} = \left(\frac{L}{l}\right)^2 = 132\cdot25.$$

$$\frac{\text{Displacement of ship in tons salt water, or cubic ft.}}{\text{Displacement of model in tons salt water, or cubic ft.}} = \left(\frac{L}{l}\right)^3 = 1\,521.$$

Multiplier for residuary resistance :—

$$\frac{\text{Residuary resistance, tons or lbs., of ship in salt water}}{\text{Residuary resistance, tons or lbs., of model in salt water}} = \left(\frac{L}{l}\right)^3 = 1\,521.$$

$$\frac{\text{Residuary resistance in lbs. of ship in salt water}}{\text{Residuary resistance in lbs. of model in fresh water}} = \frac{36}{35}\left(\frac{L}{l}\right)^3 = 1\,564\cdot5.$$

20-ft. model. Wetted surface = 72·46 sq. ft. (S) = 6·35. O = ·114 70. δ = displacement in lbs. in fresh water = 2 405 lbs.

Knots speed.	Skin frictional resistance in lbs.					(C)
	Washington tank, f = ·009 67 n = 1·854.	Tideman's, f = ·008 34 n = 1·94.	W. Froude's, f = ·0088 n = 1·94.	R. E. Froude's, f = ·010 55 n = 1·825.	R. E. Froude's, OSL - ·175.	
3	5·36	5·1	5·39	5·68		
4	9·15	8·9	9·4	9·55	9·35	1·017
5	13·81	13·7	14·32	14·41	14·01	1·311
6	19·39	19·5	20·58	19·9	19·54	2·435
6·3	...	21·45	22·62			

(C) $= \frac{r}{\delta^{\frac{2}{3}} v^2} \times 232\cdot5$, where r = total resistance, and v = hundreds of feet per minute.

TABLE V.—FOR SURFACE FRICTION OF SHIPS IN SALT WATER.

Froude's Frictional Constants for paraffin, varnish, or smooth hard surfaces—clean, painted steel,—corrected for *Salt Water*.

Length in feet.	Coefficient of friction.	Index, or power, according to which friction varies.	Length in feet.	Coefficient of friction.	Index, or power, according to which friction varies.
	f.	*n*.		*f*.	*n*.
14	·010 80	1·825	*300	·008 90	1·825
20	·010 40	,,	320	·008 886	,,
25	·010 17	,,	340	·008 872	,,
30	·010 00	,,	360	·008 857	,,
35	·009 85	,,	380	·008 844	,,
40	·009 76	,,	*400	·008 83	,,
45	·009 68	,,	420	·008 817	,,
*50	·009 6	,,	440	·008 805	,,
60	·009 47	,,	460	·008 873	,,
*75	·009 35	,,	480	·008 781	,,
90	·009 25	,,	*500	·008 77	,,
*100	·009 2	,,	520	·008 759	,,
110	·009 16	,,	540	·008 749	,,
120	·009 13	,,	560	·008 739	,,
130	·009 105	,,	580	·008 730	,,
140	·009 08	,,	600	·008 721	,,
150	·009 06	,,	620	·008 713	,,
160	·009 04	,,	640	·008 704	,,
170	·009 02	,,	660	·008 696	,,
180	·009 00	,,	680	·008 688	,,
190	·008 99	,,	*700	·008 68	,,
*200	·008 98	,,	720	·008 672	,,
210	·008 972	,,	740	·008 664	,,
220	·008 964	,,	760	·008 656	,,
230	·008 956	,,	780	·008 648	,,
240	·008 948	,,	800	·008 640	,,
250	·008 940	,,	820	·008 632	,,
260	·008 932	,,	840	·008 624	,,
270	·008 924	,,	860	·008 616	,,
280	·008 916	,,	880	·008 608	,,
290	·008 908	,,	*900	·008 6	,,

The values marked thus * are taken from Mr G. S. Baker's book, *Ship Form, Resistance, and Screw Propulsion* (Constable, 1915).

TABLE VI.—SURFACE FRICTION CONSTANTS FOR SHIPS IN SALT WATER OF 1·026 DENSITY.

(Based upon Tideman's Experiments.)

Length of Ship	Values of f for Steel Bottom Clean and Well Painted	n	Values of f for other surfaces: Clean Copper Sheets	Common Iron Skin	Smooth Sawn Plank	Moderately Foul	Barnacled
10	·01124	1·853					
15	·01098	1·851					
20	·01075	1·849					
25	·01036	1·846					
30	·01018	1·844					
35	·01006	1·842					
40	·01000	1·8397	·007	·014	·016	·019	·055
50	·00991	1·8357					
75	·00978	1·8315					
100	·00970	1·83					
150	·00957	1·83					
200	·00944	1·83					
250	·00933	1·83					
300	·00923	1·83					
350	·00916	1·83					
400	·00910	1·83					
450	·00906	1·83					
500	·00904	1·83					
550	·00901	1·83					
600	·00899	1·83					
650	·00896	1·83					
700	·00894	1·83					
750	·00892	1·83					
800	·00890	1·83					

The skin friction on p. 22, taken from our first edition, was calculated from the constants on Table II, with $n = 1{\cdot}94$ (Plate 1). In Plate 2 we have taken the values of f and n, used for 20-ft. wooden models at the U.S. tank at Washington, as a basis for a new curve following the shape of those by W. Froude and Tideman. This gives the values for tank models in fresh water found in Table IV.

Skin Resistance of Model $= fSV^n$ where S is the wetted surface,
$f =$ the coefficient of friction,
$V =$ the speed,

and n the index of variation of speed with resistance.

For values of f and n see Tables I to IV. Subtract the calculated skin resistance from the total resistance, and the remainder is the residuary resistance.

By the Law of Comparison the corresponding residuary resistance for the ship can be found.

Let l and L = length of model and vessel respectively.
v and V = corresponding speeds.
r and R = corresponding residuary resistances.

Then

$$\frac{V}{v} = \sqrt{\frac{L}{l}}$$

and

$$\frac{R}{r} = \left(\frac{L}{l}\right)^3.$$

Next, the surface friction resistance of the actual ship can be calculated, from the values of f and n in Tables V to VII, and added to the residuary resistance just determined. The sum is the total resistance of the steamer. As the model experiments are made in fresh water, and the ship has to sail in salt water, we multiply by $\frac{36}{35}$, thus $\frac{R}{r} = \frac{36}{35}\left(\frac{L}{l}\right)^3$.

Skin H.P. per 1 000 sq. ft. of wetted surface for iron or steel ships, clean and well painted. (Salt water.)

Skin H.P. $= f \times 1\,000 \times {\cdot}003\,070\,7 \times V^{2{\cdot}83}$. (Table X.)

Skin resistance per 1 000 sq. ft. $= f \times 1\,000 \times {\cdot}003\,070\,7 \times V^{1{\cdot}83}$. (Table VIII.)

TABLE VII.—FRICTIONAL CONSTANTS FOR SHIPS IN *SALT WATER*, BASED UPON TIDEMAN'S EXPERIMENTS.

Length of ship in feet.	Steel bottom clean and well painted.		Copper- or zinc-sheathed. Sheathing smooth and in good condition.		Copper- or zinc-sheathed. Sheathing rough and in bad condition.	
	f.	*n.*	*f.*	*n.*	*f.*	*n.*
* 10	·011 24	1·853 0	·010 00	1·917 5	·014 00	1·870 0
* 20	·010 57	1·848 4	·009 90	1·900 0	·013 50	1·861 0
* 30	·010 18	1·844	·009 03	1·865 0	·013 10	1·853 0
* 40	·009 98	1·839 7	·009 78	1·840 0	·012 75	1·847 0
* 50	·009 91	1·835 7	·009 76	1·830 0	·012 50	1·843 0
60	·009 86					
70	·009 81					
80	·009 77					
90	·009 73					
* 100	·009 70	1·829	·009 66	1·827 0	·012 00	1·843 0
110	·009 67	1·829		,,		,,
120	·009 64	,,		,,		,,
130	·009 61	,,		,,		,,
140	·009 59	,,		,,		,,
* 150	·009 57	1·829	·009 53	1·827 0	·011 88	1·843 0
175	·009 49	,,		,,		,,
* 200	·009 44	1·829	·009 43	1·827 0	·011 70	,,
225	·009 39	,,		,,		,,
* 250	·009 33	,,	·009 36	,,	·011 60	,,
275	·009 28	,,		,,		,,
* 300	·009 23	,,	·009 30	,,	·011 52	,,
325	·009 195	,,		,,		,,
* 350	·009 16	,,	·009 27	,,	·011 45	,,
375	·009 128	,,		,,		,,
* 400	·009 10	1·829	·009 26	,,	·011 40	,,
425	·009 077	,,		,,		,,
* 450	·009 06	,,	·009 26	,,	·011 37	,,
475	·009 05	,,		,,		,,
* 500	·009 04	1·829	·009 26	,,	·011 36	,,
550	·009 01	,,		,,		,,
600	·008 99	,,		,,		,,
650	·008 97	,,		,,		,,
700	·008 95	,,		,,		,,
750	·008 93	,,		,,		,,
800	·008 92	,,		,,		,,
850	·008 91	,,		,,		,,

1·83 is usually taken instead of 1·829.

Lines marked thus * are taken from Mr D. W. Taylor's book, *The Speed and Power of Ships* (1911).

TABLE VIII.—SKIN FRICTIONAL RESISTANCE IN LB. PER 1000 SQUARE FEET OF WETTED SURFACE FOR VARIOUS LENGTHS OF SHIPS AT DIFFERENT SPEEDS.

Speed in Knots	100 ft. $n=1\cdot83$ $f=\cdot00970$	150 ft. $n=1\cdot83$ $f=\cdot00957$	200 ft. $n=1\cdot83$ $f=\cdot00944$	300 ft. $n=1\cdot83$ $f=\cdot00923$	400 ft. $n=1\cdot83$ $f=\cdot00910$	500 ft. $n=1\cdot83$ $f=\cdot00904$	600 ft. $n=1\cdot83$ $f=\cdot00899$	700 ft. $n=1\cdot83$ $f=\cdot00894$
4·5	151·7	149·6	147·5	144·3	142·3	141·3	140·6	139·7
5·0	184·1	181·7	179·3	175·2	172·9	171·7	170·7	169·8
5·5	219·0	216·0	212·5	209·0	207·0	205·0	..	202·0
6·0	257	253·4	250	246·4	244	242	239	236·7
6·25	276·5	273·5	269	264·0	262	259	..	255·5
6·5	297·0	294·0	289	284·0	281	277·5	..	273·5
6·85	327·5	323·7	317	314	312	308	304	300
7·0	340·0	336·0	330	323·5	321	317	..	312
7·25	362·5	360·0	353·5	345·5	342	338·0	..	333·5
7·5	386·5	383·5	375·5	367·5	363·5	359·0	..	355·0
7·75	410·0	407·0	399·0	392·0	386·0	381·0	..	376·0
8·0	434·0	431·0	422·5	415·0	408·0	404·0	..	400·0
8·25	460·0	455·5	446·5	438·0	432·5	427·5	..	424·0
8·575	498·5	487	480·5	470	463	460	457	455
8·75	511·0	516·0	496·5	487	480·0	475·5	..	471·5
9·0	540·0	532·5	524·0	513	506·5	502·5	..	497·5
9·25	566·0	560·0	550·0	540·0	532·0	528·0	..	522·5
9·5	596·5	589·0	578·0	567·5	560·5	556·0	..	550·0
9·71	621·5	613·5	605	591	583·5	580	576	573
10·0	661·0	647·5	636	625	616·0	611	..	605
10·29	692	684	674	659	649	644·5	641	637
10·5	716·5	707·5	695	682·5	674	667·5	..	659·5
10·85	761	751	741	724	714	709	705	701
11·0	780	770	756	740	732	725	..	717·5
11·2	805	797·5	781·5	766·5	757·5	750	..	743·0
11·4	834	823	807	792	781	776	773	767
11·6	860	848	832	816	805·5	800	..	792·0
11·8	886·5	875	858	842·5	830·0	825	..	816·5
12·0	913	901	885	869	856	850·6	845	841
12·25	946·5	935	916·5	902	887·5	882·5	..	872·5
12·57	990	979	960	945	930	924	919	914
12·75	1015	1005	980	970	955	950	945	936
13·0	1057	1043	1029	1005	991	985	980	974
13·25	1088	1078	1062	1040	1025	1019	1010	1005
13·5	1136	1120	1100	1078	1062	1058	1052	1047
13·75	1168	1152	1140	1110	1100	1090	1085	1078

TABLE VIII.—SKIN FRICTIONAL RESISTANCE IN LB. PER 1000 SQUARE FEET OF WETTED SURFACE FOR VARIOUS LENGTHS OF SHIPS AT DIFFERENT SPEEDS—(*continued*).

Speed in Knots.	100 ft. $n=1{\cdot}83$ $f=\cdot00970$	150 ft. $n=1{\cdot}83$ $f=\cdot00957$	200 ft. $n=1{\cdot}83$ $f=\cdot00944$	300 ft. $n=1{\cdot}83$ $f=\cdot00923$	400 ft. $n=1{\cdot}83$ $f=\cdot00910$	500 ft. $n=1{\cdot}83$ $f=\cdot00904$	600 ft. $n=1{\cdot}83$ $f=\cdot00899$	700 ft. $n=1{\cdot}83$ $f=\cdot00894$
14·0	1214	1197	1180	1156	1140	1131	1125	1115
14·25	1249	1232	1217	1187	1170	1163	1157	1150
14·5	1290	1275	1258	1227	1208	1200	1193	1187
14·75	1332	1312	1298	1267	1250	1240	1232	1225
15·0	1377	1359	1340	1310	1292	1283	1276	1269
15·25	1417	1400	1380	1348	1330	1320	1310	1302
15·5	1460	1442	1422	1390	1372	1359	1350	1342
15·75	1500	1487	1464	1432	1412	1400	1390	1382
16·0	1550	1530	1507	1476	1452	1441	1433	1425
16·25	1590	1570	1555	1519	1500	1582	1571	1462
16·5	1638	1620	1598	1560	1540	1527	1513	1502
16·75	1680	1665	1642	1602	1582	1568	1560	1545
17·0	1726	1710	1682	1648	1623	1612	1602	1593
17·25	1774	1759	1734	1694	1670	1658	1643	1636
17·5	1822	1802	1780	1738	1717	1705	1687	1680
17·75	1870	1852	1825	1781	1760	1647	1631	1622
18·0	1923	1900	1873	1833	1805	1793	1780	1770
18·25	1970	1947	1920	1878	1854	1840	1822	1812
18·5	2022	2000	1970	1920	1900	1890	1870	1860
18·75	2070	2048	2019	1970	1947	1937	1915	1904
19·0	2122	2095	2067	2020	1994	1980	1970	1956
19·25	2172	2150	2118	2067	2045	2028	2012	2000
19·5	2226	2200	2166	2117	2097	2080	2061	2050
19·75	2279	2250	2218	2167	2142	2124	2108	2097
20·0	2331	2303	2270	2220	2188	2174	2160	2150
20·25	2386	2357	2320	2268	2242	2222	2206	2195
20·5	2440	2408	2378	2320	2293	2261	2255	2244
20·75	2492	2460	2425	2372	2343	2325	2307	2296
21·0	2550	2517	2480	2427	2391	2375	2361	2350
21·25	2600	2572	2532	2478	2447	2425	2408	2398
21·5	2658	2626	2588	2530	2500	2480	2460	2450
21·75	2712	2682	2642	2585	2550	2533	2517	2503
22·0	2772	2739	2700	2640	2605	2583	2570	2555
22·25	2830	2798	2755	2700	2660	2640	2623	2608
22·5	2888	2856	2810	2750	2713	2693	2678	2663
22·75	2945	2910	2867	2809	2770	2747	2730	2715

TABLE VIII.—SKIN FRICTIONAL RESISTANCE IN LB. PER 1000 SQUARE FEET OF WETTED SURFACE FOR VARIOUS LENGTHS OF SHIPS AT DIFFERENT SPEEDS—(*continued*).

Speed in Knots	100 ft. $f = \cdot00970$ $n = 1\cdot83$	150 ft. $f = \cdot00957$ $n = 1\cdot83$	200 ft. $f = \cdot00944$ $n = 1\cdot83$	300 ft. $f = \cdot00923$ $n = 1\cdot83$	400 ft. $f = \cdot00910$ $n = 1\cdot83$	500 ft. $f = \cdot00904$ $n = 1\cdot83$	600 ft. $f = \cdot00899$ $n = 1\cdot83$	700 ft. $f = \cdot00894$ $n = 1\cdot83$
23·0	3005	2970	2926	2865	2825	2805	2792	2775
23·25	3065	3028	2982	2925	2882	2860	2845	2830
23·5	3127	3090	3040	2980	2940	2918	2900	2885
23·75	3187	3150	3105	3090	2990	2974	2860	3941
24·0	3255	3210	3164	3100	3054	3030	3020	3000
24·25	3314	3270	3225	3160	3115	3088	3078	3060
24·5	3378	3332	3288	3220	3172	3145	3133	3120
24·75	3440	3395	3350	3280	3232	3208	3192	3180
25·0	3503	3460	3400	3340	3280	3260	3244	3230
25·25	3568	3520	3475	3405	3354	3326	3314	3300
25·5	3632	3587	3538	3463	3412	3388	3376	3362
25·75	3700	3652	3600	3528	3478	3450	3435	3420
26·0	3770	3720	3664	3590	3540	3510	3496	3475
26·25	3832	3780	3727	3654	3600	3570	3558	3540
26·5	3900	3842	3792	3720	3660	3630	3618	3600
26·75	3970	3910	3858	3780	3725	3695	3680	3660
27·0	4040	3980	3925	3845	3790	3760	3740	3720
27·25	4105	4048	3987	3908	3850	3822	3802	3782
27·5	4175	4110	4053	3970	3910	3882	3862	3842
27·75	4240	4180	4120	4040	3980	3950	3930	3910
28·0	4312	4253	4195	4105	4045	4015	3990	3972
28·25	4380	4320	4257	4170	4110	4075	4060	4040
28·5	4450	4385	4325	4240	4180	4140	4120	4100
28·75	4522	4460	4398	4307	4242	4210	4188	4168
29·0	4595	4540	4460	4373	4310	4280	4255	4240
29·25	4667	4603	4535	4440	4380	4342	4320	4300
29·5	4738	4672	4600	4510	4450	4410	4388	4368
29·75	4810	4747	4677	4580	4520	4480	4458	4438
30·0	4880	4840	4755	4652	4590	4550	4530	4500
30·25	4960	4887	4820	4720	4660	4620	4598	4578
30·5	5020	4960	4890	4790	4740	4690	4660	4640
30·75	5120	5050	4970	4860	4795	4762	4735	4695
31·0	5200	5135	5050	4950	4880	4850	4815	4780
31·25	5265	5205	5125	5010	4940	4910	4880	4840
31·5	5345	5285	5204	5085	5015	4985	4953	4912
31·75	5430	5365	5283	5165	5095	5060	5030	4992
32·0	5510	5454	5350	5250	5167	5140	5104	5075

Table IX.—Skin Horse-Power per 1000 Square Feet of Wetted Surface for Various Lengths of Ships at Different Speeds (from Curves).

Speed in Knots	100 ft.	150 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	700 ft.
4·5	2·09	2·062	2·038	1·99	1·965	1·95	1·94	1·927
4·75	2·44	2·4	2·379	2·343	2·315	2·29	2·28	2·27
5·00	2 83	2·79	2·75	2·69	2·655	2·63	2·62	2·615
5·25	3·253	3·215	3·19	3·15	3·1	3·08	3·042	3·01
5·50	3·68	3·63	3·57	3·51	3·48	3·44	3·41	3·39
5·75	4·21	4·14	4·10	4·03	3·99	3·95	3·945	3·94
6·00	4·74	4·66	4·60	4·55	4·50	4·46	4·40	4·36
6·25	5·36	5·27	5·2	5·11	5·07	5·015	4·962	4·92
6·5	5·93	5·86	5·77	5·67	5·61	5·55	5·51	5.46
6·75	6·62	6·52	6·335	6·2	6·15	6·10	6·05	5·99
7·00	7·31	7·225	7·1	6·95	6·9	6·81	6·76	6·71
7·25	8·1	8·0	7·84	7·68	7·6	7·52	7·48	7·43
7·50	8·875	8·8	8·61	8·45	8·35	8·26	8·21	8·16
7·75	8·88	8·99	9·08	9·2	9·38	9·56	9·6	9·65
8·00	10·7	10·61	10·4	10·25	10·05	9·95	9·9	9·85
8·25	11·75	11·625	11·425	11·25	11·05	10·925	10·875	10·825
8·5	12·8	12·65	12·45	12·225	12·05	11·90	11·85	11·80
8·75	13·85	13·675	13·475	13·225	13·025	12·9	12·82	12·75
9·00	14·9	14·7	14·45	14·2	14·0	13·9	13·82	13·75
9·25	16·3	16·03	15·8	15·5	15·3	15·1	14·98	14·95
9·5	17·6	17·3	17·03	16·65	16·49	16·3	16·22	16·15
9·75	18·85	18·6	18·32	17·92	17·75	17·54	17·44	17·34
10·00	20·22	19·9	19·55	19·2	18·95	18·8	18·7	18·6
10·25	21·45	21·2	20·85	20·42	20·25	19·98	19·9	19·82
10·5	22·95	22·7	22·3	21·9	21·7	21·45	21·35	21·25
10·75	24·575	24·2	23·8	23·4	23·1	22·82	22·72	22·62
11·00	26·25	25·95	25·45	24·95	24·65	24·4	24·3	24·2
11·25	27·9	27·4	26·95	26·4	26·1	25·9	25·8	25·7
11·5	29·65	28·94	28·45	27·95	27·6	27·3	27·2	27·1
11·75	31·4	30·6	30·15	29·7	29·3	29·13	28·97	28·8
12·00	33·6	33·2	32·6	32·0	31·9	31·4	31·1	30·95
12 25	35·41	34·9	34·35	33·75	33·45	33·15	32·75	32·6
12·5	37·25	36·73	36·15	35 45	35·15	34·8	34·55	34·3
12·75	39·25	38·75	38·05	37·35	37·00	36·6	36·4	36·05
13·00	41·15	40·75	40·00	39·2	38·6	38·4	38·2	38·0
13·25	43·92	43·4	42·6	41·8	41·5	40·75	40·45	40·25
13·5	46·4	45·8	45·05	44·05	43·4	43·00	42·8	42·5

Table IX.—Skin Horse-Power per 1000 Square Feet of Wetted Surface for various Lengths of Ships at different Speeds (from Curves)—(*continued*).

Speed in Knots	100 ft.	150 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	700 ft.
13·75	49·15	48·4	47·7	46·8	46·0	45·6	45·4	45·05
14·00	52·3	51·5	50·75	49·75	49·00	48·6	48·4	48·00
14·25	55·2	54·35	53·6	52·6	51·8	51·45	51·1	50·8
14·5	57·75	57·05	56·35	55·2	54·4	54·0	53·65	53·4
14·75	60·5	59·8	59·05	57·75	56·75	56·6	56·25	56·0
15·00	63·5	62·6	61·9	60·5	59·6	59·3	58·9	58·5
15·25	66·8	66·0	64·82	63·6	62·42	62·3	61·8	61·4
15·5	69·6	69·0	67·75	66·2	65·4	65·1	64·70	64·3
15·75	73·0	72·2	70·8	69·2	68·4	68·1	67·5	67·1
16·00	76·25	75·25	74·1	72·5	71·5	71·0	70·5	70·1
16·25	79·8	78·8	77·6	75·8	74·9	74·4	74·0	73·5
16·5	83·4	82·6	81·2	79·4	78·4	78·0	77·4	76·8
16·75	87·0	86·15	84·6	82·8	81·8	81·4	80·95	80·2
17·00	90·5	89·5	88·0	86·15	85·00	84·4	83·75	83·35
17·25	94·54	93·4	92·15	90·1	88·9	88·4	87·75	87·1
17·50	98·4	97·5	96·0	93·9	92·7	92·25	91·55	90·9
17·75	102·25	101·2	100·0	97·75	96·4	95·9	95·2	94·6
18·00	106·3	105·0	103·85	101·4	100·0	99·4	98·6	98·1
18·25	110·4	109·8	107·5	105·8	103·6	102·8	102	101·4
18·5	114·15	112·4	111·1	108·4	107·2	106·15	105·4	104·8
18·75	118·15	116·2	114·8	112·0	110·6	109·8	108·95	108·3
19·00	122	120	118·8	116·0	114·5	113·75	112·9	112·02
19·25	127·4	125·35	124·0	121·1	119·5	118·7	117·7	116·9
19·5	132·1	130·3	128·7	127·6	124·0	123·1	122·2	121·2
19·75	137·75	136·5	134·6	131·4	129·7	128·8	127·8	126·8
20·00	143·2	141·8	139·5	136·5	134·65	134·0	133·0	132·1
20·25	148·3	147·75	144·5	141·5	139·3	138·8	138·0	137·0
20·5	153·3	151·5	149·0	146·2	144·0	143·25	142·5	141·4
20·75	158·5	156·7	154·1	151·0	148·5	148·0	147·2	146·4
21·00	164·5	162·0	160·0	156·8	154·5	153·1	152·5	151·8
21·25	170·0	167·5	165·1	161·75	159·3	158·0	157·25	156·5
21·5	175·5	172·75	170·25	167·00	164·7	163·0	162·25	161·5
21·75	180·5	178·75	176·3	172·5	170·5	169·0	168·25	167·1
22·00	187·5	185·0	182·5	178·5	176·5	174·75	174·0	172·5
22·25	193·4	190·5	188·0	184·25	182·25	180·5	179·5	178·25
22·50	199·5	196·25	194·0	190·25	187·85	186·35	185·5	184·3
22·75	206·0	202·75	201·4	196·0	193·7	192·25	191·7	190·15

Table IX.—Skin Horse-Power per 1000 Square Feet of Wetted Surface for various Lengths of Ships at different Speeds (from Curves)—(*continued*).

Speed in Knots	100 ft.	150 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	700 ft.
23·00	212·25	210·00	207·0	202·3	200·0	198·5	197·6	196·0
23·25	219·0	215·75	213·3	208·75	206·0	204·5	203·75	202·0
23·50	225·8	222·8	220·0	205·25	212·4	211·0	210·0	208·4
23·75	232·75	229·5	227·0	222·0	219·0	217·25	216·5	215·0
24·00	239·75	236·5	233·5	228·25	225·0	223·5	222·5	221·0
24·25	247·75	244·0	240·5	235·2	232·0	230·2	229·2	227·5
24·5	255·00	251·25	247·15	242·0	238·75	237·0	236·0	234·0
24·75	262·5	258·1	254·1	249·0	245·0	243·4	242·5	240·5
25·00	269·5	266·0	261·5	256·5	252·0	250·5	249·5	248·0
25·25	277·5	273·5	269·15	264·1	260·0	258·5	257·5	256·0
25·5	285·75	281·25	276·3	271·3	267·3	266·0	265·0	263·3
25·75	293·5	289·25	284·0	279·25	274·4	273·25	272·0	270·75
26·00	301·9	297·5	293·5	287·0	282·5	281·0	279·5	278·0
26·25	310·5	305·8	300·9	294·5	290·0	288·75	287·0	285·6
26·5	319·0	314·25	309·75	303·0	298·8	296·8	285·25	283·8
26·75	327·5	322·0	317·5	310·5	306·0	304·0	302·0	300·5
27·00	336·00	330·0	326·0	319	314·5	312·0	310·0	308·25
27·25	342·75	337·5	333·0	326·0	321·5	319·0	317·0	315·0
27·5	350·0	345·0	340·0	333·25	328·0	325·5	324·0	321·5
27·75	357·75	352·5	347·5	341·0	336·0	333·0	331·25	329·0
28·00	366·25	361·0	355·0	348·5	343·25	340	338·3	336·0
28·25	375·00	369·25	363·5	356·5	351·25	348·0	346·5	344·25
28·5	384·0	378·5	372·5	366·0	360·5	357·0	355·5	353·0
28·75	394·5	389·25	382·5	376·0	370·0	366·5	365·0	363·0
29·00	405·0	400·0	393·0	385·0	380·0	376·0	374·0	372·0
29·25	416·0	410·5	404·0	397·0	391·0	387·0	385·0	382·5
29·5	427·5	423·0	415·5	408·0	402·0	398·0	396·0	394·0
29·75	438·5	434·25	426·5	418·5	413·0	408·5	406·5	404·0
30·00	451·0	446·0	439·0	430·0	425·0	420·0	418·0	416·0
30·25	461·5	456·3	449·5	440·5	434·5	431·5	428·5	426·5
30·50	472·5	466·75	459·5	451·0	444·4	442·0	438·8	436·25
30·75	433·4	477·2	470·2	461·3	455·0	453·0	449·0	446·5
31·00	495·0	488·0	480·0	471·5	465·0	462·0	459·0	456·0
31·25	506·0	498·75	491·9	483·0	476·0	474·0	470·0	467·2
31·50	518·0	510·5	503·0	490·3	487·0	484·0	480·5	470·7
31·75	530·0	522·0	514·4	500·75	480·0	494·75	491·5	488·5
32·00	544·0	536·0	527·0	517·0	510·0	506	502·5	500·0

TABLE X.

	Skin H.P. from Froude's constants for salt water: f = ·008 92 for 300 ft. f = ·009 03 for 188 ft. n taken at 2·83.	Skin H.P. from our tables, based on Tideman's constants for salt water: f = ·009 23 for 300 ft. f = ·009 46 for 188 ft. n = 2·83.
H.M.S. "Iris," 18·573 knots, 300 × 46·08 × 18·08. Displacement = 3 290 tons. Mid-area coefficient = ·889. Wetted surface by Mumford's (Denny's) formula = 15 570 sq. ft., with an addition of 5 per cent., making wetted surface = 16 340	1 739	1 833
H.M.S. "Iris," 18·573 knots, same dimensions, but wetted surface taken from Mr G. S. Baker's book = 18 600 sq. ft., which is 19½ per cent. over the value given by Denny's formula (and possibly includes appendages)	1 980	2 048
U.S.S. "Manning," 188 × 32·81 × 12·33 ft. mean draught. Δ = 1 000·7 tons. 16 knots. Wetted surface given by Prof. Peabody as 7 273 sq. ft., which is 5½ per cent. above the value calculated from Denny's formula	515	539
T.S.S. "H," 418 × 52 × 23 ft. mean draught. Δ = 9 100 tons. Block coefficient = ·637. Mid-area coefficient = ·956. Wetted surface from Denny's formula = 30 300 sq. ft. 14½ knots	1 592	1 635

Other methods of arriving at the skin friction horse-power are the following:—

(1) Mr R. E. Froude's $F_M - F_S = (O_M - O_S)SL^{-\cdot175}$, using the values of O in the table (p. 78). This is the method employed at Haslar, and is the basis of the correction for (c) value used by Mr Baker at the National Physical Laboratory.

(2) Mr D. W. Taylor's Contours of Frictional Resistance in

pounds per ton of displacement, the ordinates being Displacement length coefficient $\dfrac{D}{\left(\dfrac{L}{100}\right)^3}$, up to 160, and the abscissæ Speed-length-ratio $\dfrac{V}{\sqrt{L}}$.

See *The Speed and Power of Ships*, vol. ii, fig. 78.

(3) Tables VIII and IX in this book, giving skin H.P. and resistance per 1 000 sq. ft. of wetted surface at various speeds, perhaps the handiest fo r naval architects engaged in ordinary work.

In a paper read before the Institution of Naval Architects in April 1916, Mr G. S. Baker gave an account of experiments made recently at the National Physical Laboratory, and also by Beaufoy, which showed that the skin friction of a ship-shaped form was considerably in excess of that of a plane board. The ordinary method of calculating the skin frictional resistance of a model or ship is based upon the hypothesis that the immersed skin is equivalent in resistance to that of a rectangular plane surface of equal area and length in line of motion, but Mr Baker's experiments at the National Physical Laboratory, with models towed at very low speeds, showed that the resistances of all the models were in excess of those for planks of the same wetted surface, and that the fulness of the form affected the result. Beaufoy tested several submerged to such a depth that wave-making was absent. Dr Lees advocated towing submarines of 100 ft. to 200 ft. in length, and it is hoped that this will be found possible. Mr Baker's experiments gave the following results :—

TABLE XI.

Type of model.	Length in feet.	Prismatic coefficient.	$\dfrac{\text{Actual skin resistance}}{\text{Calculated skin resistance}}$.	
Mercantile steamer .	16·0	·60	1·1	
T.B. destroyer . .	14·4	·64	1 05	
Battleship . .	14·4	·63	1·1	
"Greyhound" . .	10·8	·68	1·1	
Mercantile steamer .	16·0	·68	1·11	
,, .	15·9	·69	1·14	
,, .	16·0	·70	1·19	
,, .	16·0	·76	1·17	
,, .	15·0	·81	1·23	Some eddy-making present
,, .	16·0	·83	1·29	

Using Captain Dyson's figures, we have the following :—

Name.	Beam as percentage of length.	⊗ Coef.	Block coef.	Prismatic coef.	Appendage resistance in percentage of bare hull resistance.	No. of shafts.
Baltimore . .	15·4	·842	·515	·612	12·1	2
Biddle . .	11·2	·724	·478	·660	9·7	2
Birmingham .	11·2	·667	·405	·608	11·0	2
Castine . .	15·7	·854	·504	·590	12·8	2
Chester . .	11·2	·724	·400	·553	11·3	4
Cincinnati .	14·0	·873	·493	·565	13·4	2
Columbia . .	14·1	·869	·491	·566	13·2	3
Cushing . .	10·4	·700	·386	·552	10·9	2
Cyclops . .	12·3	·984	·726	·739	12·9	2
Decatur . .	9·4	·658	·461	·702	8·3	2
Delaware . .	16·7	·978	·600	·614	18·0	2
50-ft. launch .	20·0	...	·352	...	2·7	1
Fuel barge .	15·6	·980	·886	·904	3·6	1
Indiana . .	19·9	·931	·622	·669	16·1	2
Iowa . .	20·06	·944	·630	·668	19·4	2
Katahdin . .	16·7	·734	·461	·629	7·0	2
Kentucky . .	19·6	·957	·643	·672	18·7	2
Macdonough .	9·2	·755	·404	·535	12·1	2
Mackenzie .	12·9	·700	·420	·600	2·3	1
Maine (old) .	17·9	·859	·574	·669	12·2	2
Monterey . .	23·1	·905	·643	·710	15·4	2
New Jersey .	17·5	·906	·656	·724	13·4	2
North Dakota .	16·7	·978	·600	·614	17·0	2
Orion . .	12·5	·986	·726	·736	12·9	2
Paducah . .	20·0	·860	·520	·605	13·7	2
Preble . .	9·6	·770	·410	·533	30·0	2
Smith . .	9·0	·649	·407	·628	9·7	3
Stockton . .	10·0	·730	·399	·547	11·8	2
Talbot . .	12·6	·800	·337	·421	3·6	1
Truxtun . .	9·0	·675	·370	·549	10·9	2
Utah . .	17·3	·979 2	·583 7	·596	15·8	4
Vicksburg .	21·4	·820	·482	·589	3·0	1
Wyoming . .	16·8	·986	·618	·626	15·4	4

See also p. 377.

CHAPTER III.

THE LAW OF COMPARISON OR PRINCIPLE OF SIMILITUDE.

Ratios used in applying the Law of Comparison when passing from one size of ship to another, at corresponding speeds.

LET us call any ship whose residuary resistance, or residuary or wave-making effective horse-power is known, the type ship; then, if we are considering another vessel l times as long as the type ship $\left(i.e.\ \frac{L_1}{L} = l\right)$,*

All linear dimensions vary as	l
Speeds of ship, speeds of revolution, etc., vary as	$\sqrt{l}$
Surfaces, wetted skin of ship, midship areas, piston areas, etc., vary as	l^2
Displacements, weights, and cubic measurements vary as .	l^3
Pressures in engines, water or steam, and residuary resistances, thrust and torque, vary as	l^3
Residuary horse-powers vary as $\sqrt{l} \times l^3$, *i.e.*	$l^{3 \cdot 5}$

These simple mathematical ratios, however, are not applicable to the skin friction element of the total horse-power.

$$\text{E.H.P.} = \text{residuary H.P.} + \text{skin H.P.}$$

Mr Hillhouse mentions $l^{3 \cdot 415}$ as a convenient ratio according to which skin friction power varies, deduced from Froude's and Tideman's experiments with planes towed through water.

* Professor Archibald Barr's admirable paper on "Similar Structures and Machines," read before the Institution of Engineers and Shipbuilders in Scotland in 1900, will be found interesting in this connection.

TABLE XII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON, AND CONVERTING TO 100-FT. MODELS.

Ship length L.	$\sqrt{L}$.	L^2.	L^3.	$\frac{L}{100}$ or l.	$\left(\frac{L}{100}\right)^3$ or l^3.	$\left(\frac{L}{100}\right)^{3 \cdot 5}$ or $l^{3 \cdot 5}$.
8	2·828	64	256	·08	·000 512	·000 144 79
9	3·00	81	729	·09	·000 729	·000 218 7
10	3·162	100	1 000	·10	·001 00	·000 316 2
11	3·316 6	121	1 331	·11	·001 331	000 441 4
12	3·464	144	1 728	·12	·001 728	·000 598 5
13	3·605 5	169	2 197	·13	·002 197	·000 792 1
14	3·741 6	196	2 744	·14	·002 744	·001 028
15	3·872 9	225	3 375	·15	·003 375	·001 309
16	4·00	256	4 096	·16	·004 096	·001 638 4
17	4·123 1	289	4 913	·17	·004 913	·002 024
18	4·242 6	324	5 832	·18	·005 832	·002 474
19	4·358 8	361	6 859	·19	·006 859	·002 99
20	4·472 1	400	8 000	·20	·008 00	·003 58
21	4·582 5	441	9 261	·21	·009 261	·004 24
22	4·690 4	484	10 648	·22	·010 648	·004 99
23	4·795 8	529	12 167	·23	·012 167	·005 83
24	4·898	576	13 824	·24	·013 824	·006 76

If we continued this table, the values of L^2 and L^3 would become inconveniently large, therefore we make a table of functions of l, thus:

Ship length L.	l.	$\sqrt{l}$.	l^2.	l^3.	$l^{3 \cdot 5}$.
23	·23	·479 58	·052 9	·012 167	·005 83
24	·24	·489 8	·057 6	·013 824	·006 76

$$l = \frac{\text{length of ship}}{100}.$$

and continue as in Table XIII.

Table XIII.—Multipliers Used in Applying the Law of Comparison.

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$
						60	·60	·774	·360	·2160	·1670
						61	·61	·781	·372	·2270	·1771
						62	·62	·787	·384	·2383	·1875
						63	·63	·793	·397	·2500	·1981
24	·24	·490	·0576	·0138	·00676	**64**	·64	·800	·409	·2621	·2096
25	·25	·500	·0625	·0156	·00780	**65**	·65	·806	·422	·2746	·2215
26	·26	·510	·0676	·0175	·00892	**66**	·66	·812	·435	·2875	·2335
27	·27	·519	·0729	·0197	·01023	**67**	·67	·818	·449	·3007	·2460
28	·28	·529	·0784	·0219	·01158	**68**	·68	·824	·462	·3144	·2590
29	·29	·538	·0841	·0244	·01311	**69**	·69	·830	·476	·3285	·2728
30	·30	·547	·0900	·0270	·01476	**70**	·70	·836	·490	·3430	·2870
31	·31	·556	·0961	·0298	·01658	**71**	·71	·842	·504	·3579	·3012
32	·32	·565	·1024	·0327	·01849	**72**	·72	·848	·518	·3732	·3160
33	·33	·574	·109	·0359	·0206	**73**	·73	·854	·533	·3890	·3320
34	·34	·583	·115	·0393	·0229	**74**	·74	·860	·547	·4052	·348
35	·35	·591	·122	·0428	·0253	**75**	·75	·866	·562	·4218	·365
36	·36	·600	·129	·0466	·0279	**76**	·76	·871	·577	·4389	·382
37	·37	·608	·137	·0506	·03075	**77**	·77	·877	·593	·4565	·400
38	·38	·616	·144	·0548	·03371	**78**	·78	·883	·608	·4745	·418
39	·39	·624	·152	·0593	·0370	**79**	·79	·889	·624	·4930	·438
40	·40	·632	·160	·0640	·0404	**80**	·80	·891	·640	·5120	·456
41	·41	·640	·168	·0690	·04415	**81**	·81	·900	·656	·5314	·478
42	·42	·648	·176	·0741	·0480	**82**	·82	·905	·672	·5513	·499
43	·43	·655	·185	·0795	·0521	**83**	·83	·911	·690	·5717	·521
44	·44	·663	·193	·0852	·05645	**84**	·84	·916	·705	·5927	·543
45	·45	·671	·202	·0911	·0611	**85**	·85	·922	·722	·6141	·566
46	·46	·678	·211	·0973	·0659	**86**	·86	·927	·739	·6360	·590
47	·47	·685	·221	·1038	·0711	**87**	·87	·932	·757	·6585	·613
48	·48	·693	·230	·1105	·0765	**88**	·88	·938	·774	·6814	·639
49	·49	·700	·240	·1176	·0823	**89**	·89	·943	·792	·7049	·665
50	·50	·707	·250	·1250	·0883	**90**	·90	·948	·810	·7290	·691
51	·51	·714	·260	·1326	·0946	**91**	·91	·954	·828	·7535	·718
52	·52	·721	·270	·1406	·1014	**92**	·92	·959	·846	·7786	·746
53	·53	·728	·281	·1488	·1082	**93**	·93	·964	·865	·8043	·775
54	·54	·734	·291	·1574	·1155	**94**	·94	·969	·883	·8306	·805
55	·55	·741	·302	·1663	·1233	**95**	·95	·974	·902	·8573	·835
56	·56	·748	·313	·1756	·1312	**96**	·96	·979	·921	·8847	·865
57	·57	·755	·325	·1852	·1399	**97**	·97	·984	·941	·9126	·897
58	·58	·761	·336	·1951	·1485	**98**	·98	·990	·960	·9411	·932
59	·59	·768	·348	·2053	·1578	**99**	·99	·995	·980	·9703	·965

TABLE XIII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON— (*continued*).

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3\cdot5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3\cdot5}$
100	1·00	1·00	1·00	1·00	1·00	**140**	1·40	1·183	1·960	2·744	3·245
101	1·01	1·005	1·020	1·030	1·035	**141**	1·41	1·187	1·988	2·803	3·329
102	1·02	1·010	1·040	1·061	1·070	**142**	1·42	1·191	2·016	2·863	3·410
103	1·03	1·015	1·061	1·092	1·110	**143**	1·43	1·196	2·045	2·924	3·493
104	1·04	1·019	1·081	1·124	1·145	**144**	1·44	1·20	2·073	2·986	3·580
105	1·05	1·024	1·102	1·157	1·185	**145**	1·45	1·204	2·102	3·048	3·662
106	1·06	1·029	1·123	1·191	1·226	**146**	1·46	1·208	2·131	3·112	3·755
107	1·07	1·034	1·145	1·225	1·267	**147**	1·47	1·212	2·161	3·176	3·850
108	1·08	1·039	1·166	1·259	1·308	**148**	1·48	1·216	2·190	3·241	3·940
109	1·09	1·044	1·188	1·295	1·352	**149**	1·49	1·220	2·220	3·308	4·030
110	1·10	1·048	1·210	1·331	1·396	**150**	1·50	1·224	2·250	3·375	4·130
111	1·11	1·053	1·232	1·367	1·439	**151**	1·51	1·229	2·280	3·443	4·230
112	1·12	1·058	1·254	1·405	1·487	**152**	1·52	1·232	2·310	3·511	4·328
113	1·13	1·063	1·277	1·442	1·532	**153**	1·53	1·237	2·341	3·581	4·430
114	1·14	1·067	1·299	1·481	1·580	**154**	1·54	1·241	2·371	3·652	4·530
115	1·15	1·072	1·322	1·521	1·630	**155**	1·55	1·245	2·402	3·723	4·637
116	1·16	1·077	1·345	1·560	1·680	**156**	1·56	1·249	2·433	3·796	4·740
117	1·17	1·081	1·369	1·601	1·730	**157**	1·57	1·253	2·465	3·869	4·841
118	1·18	1·086	1·392	1·643	1·784	**158**	1·58	1·257	2·496	3·944	4·960
119	1·19	1·090	1·416	1·685	1·837	**159**	1·59	1·261	2·528	4·019	5·061
120	1·20	1·095	1·440	1·728	1·890	**160**	1·60	1·265	2·560	4·096	5·180
121	1·21	1·10	1·464	1·771	1·946	**161**	1·61	1·269	2·592	4·173	5·290
122	1·22	1·104	1·488	1·815	2·005	**162**	1·62	1·272	2·624	4·251	5·410
123	1·23	1·109	1·513	1·861	2·067	**163**	1·63	1·276	2·657	4·330	5·520
124	1·24	1·113	1·537	1·906	2·120	**164**	1·64	1·280	2·689	4·411	5·645
125	1·25	1·118	1·562	1·953	2·182	**165**	1·65	1·284	2·722	4·492	5·770
126	1·26	1·122	1·587	2·000	2·245	**166**	1·66	1·288	2·755	4·574	5·890
127	1·27	1·127	1·613	2·048	2·308	**167**	1·67	1·292	2·789	4·657	6·015
128	1·28	1·131	1·638	2·097	2·370	**168**	1·68	1·296	2·822	4·741	6·143
129	1·29	1·135	1·664	2·146	2·435	**169**	1·69	1·300	2·856	4·826	6·270
130	1·30	1·140	1·690	2·197	2·502	**170**	1·70	1·304	2·890	4·913	6·407
131	1·31	1·144	1·716	2·248	2·569	**171**	1·71	1·307	2·924	5·000	6·537
132	1·32	1·149	1·742	2·299	2·641	**172**	1·72	1·311	2·958	5·088	6·668
133	1·33	1·153	1·769	2·352	2·710	**173**	1·73	1·315	2·993	5·177	6·800
134	1·34	1·157	1·795	2·406	2·785	**174**	1·74	1·319	3·027	5·268	6·940
135	1·35	1·162	1·822	2·460	2·856	**175**	1·75	1·323	3·062	5·359	7·090
136	1·36	1·166	1·849	2·515	2·930	**176**	1·76	1·326	3·097	5·451	7·23
137	1·37	1·170	1·877	2·571	3·008	**177**	1·77	1·330	3·133	5·545	7·38
138	1·38	1·174	1·904	2·628	3·085	**178**	1·78	1·334	3·168	5·639	7·52
139	1·39	1·179	1·932	2·685	3·161	**179**	1·79	1·338	3·204	5·735	7·66

TABLE XIII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON—(*continued*).

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3\cdot5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3\cdot5}$
180	1·80	1·341	3·240	5·832	7·82	**220**	2·20	1·483	4·840	10·64	15·76
181	1·81	1·345	3·276	5·929	7·97	**221**	2·21	1·486	4·884	10·79	16·04
182	1·82	1·349	3·312	6·028	8·12	**222**	2·22	1·489	4·928	10·94	16·27
183	1·83	1·352	3·349	6·128	8·28	**223**	2·23	1·493	4·973	11·09	16·54
184	1·84	1·356	3·385	6·229	8·44	**224**	2·24	1·496	5·017	11·24	16·80
185	1·85	1·360	3·422	6·331	8·61	**225**	2·25	1·500	5·062	11·39	17·08
186	1·86	1·363	3·459	6·434	8·77	**226**	2·26	1·503	5·107	11·54	17·34
187	1·87	1·367	3·497	6·539	8·94	**227**	2·27	1·506	5·153	11·69	17·60
188	1·88	1·371	3·534	6·644	9·10	**228**	2·28	1·509	5·198	11·85	17·87
189	1·89	1·374	3·572	6·751	9·28	**229**	2·29	1·513	5·244	12·00	18·14
190	1·90	1·378	3·610	6·859	9·45	**230**	2·30	1·516	5·290	12·16	18·40
191	1·91	1·382	3·648	6·967	9·63	**231**	2·31	1·519	5·336	12·32	18·70
192	1·92	1·385	3·686	7·077	9·80	**232**	2·32	1·523	5·382	12·48	19·00
193	1·93	1·389	3·725	7·189	9·98	**233**	2·33	1·526	5·428	12·65	19·30
194	1·94	1·393	3·763	7·301	10·17	**234**	2·34	1·529	5·475	12·81	19·58
195	1·95	1·396	3·802	7·415	10·35	**235**	2·35	1·533	5·522	12·97	19·87
196	1·96	1·400	3·841	7·529	10·54	**236**	2·36	1·536	5·569	13·14	20·17
197	1·97	1·403	3·881	7·645	10·73	**237**	2·37	1·539	5·617	13·31	20·50
198	1·98	1·407	3·920	7·762	10·92	**238**	2·38	1·542	5·664	13·48	20·80
199	1·99	1·410	3·960	7·880	11·10	**239**	2·39	1·546	5·712	13·65	21·10
200	2·00	1·414	4·00	8·000	11·31	**240**	2·40	1·549	5·760	13·82	21·40
201	2·01	1·417	4·040	8·120	11·50	**241**	2·41	1·552	5·808	13·99	21·70
202	2·02	1·421	4·080	8·242	11·71	**242**	2·42	1·555	5·856	14·17	22·05
203	2·03	1·424	4·120	8·365	11·91	**243**	2·43	1·558	5·905	14·34	22·34
204	2·04	1·428	4·161	8·489	12·11	**244**	2·44	1·562	5·953	14·52	22·68
205	2·05	1·431	4·202	8·615	12·33	**245**	2·45	1·565	6·002	14·70	23·01
206	2·06	1·435	4·243	8·741	12·53	**246**	2·46	1·568	6·051	14·88	23·30
207	2·07	1·438	4·285	8·869	12·76	**247**	2·47	1·571	6·101	15·07	23·67
208	2·08	1·442	4·326	8·999	12·96	**248**	2·48	1·574	6·150	15·25	24·00
209	2·09	1·445	4·368	9·129	13·18	**249**	2·49	1·577	6·200	15·43	24·35
210	2·10	1·449	4·410	9·261	13·40	**250**	2·50	1·581	6·250	15·62	24·70
211	2·11	1·452	4·452	9·393	13·62	**251**	2·51	1·584	6·300	15·81	25·04
212	2·12	1·456	4·494	9·528	13·85	**252**	2·52	1·587	6·350	16·00	25·4
213	2·13	1·459	4·537	9·663	14·09	**253**	2·53	1·590	6·401	16·19	25·7
214	2·14	1·463	4·579	9·800	14·34	**254**	2·54	1·593	6·451	16·38	26·1
215	2·15	1·466	4·622	9·938	14·56	**255**	2·55	1·596	6·502	16·58	26·5
216	2·16	1·469	4·665	10·07	14·80	**256**	2·56	1·60	6·553	16·77	26·8
217	2·17	1·473	4·709	10·21	15·05	**257**	2·57	1·603	6·605	16·97	27·2
218	2·18	1·476	4·752	10·36	15·28	**258**	2·58	1·606	6·656	17·17	27·55
219	2·19	1·479	4·796	10·50	15·53	**259**	2·59	1·609	6·708	17·37	27·9

TABLE XIII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON—(*continued*).

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$
260	2·60	1·612	6·760	17·57	28·3	**300**	3·00	1·732	9·000	27·00	46·8
261	2·61	1·615	6·812	17·77	28·7	**301**	3·01	1·735	9·060	27·27	47·3
262	2·62	1·618	6·864	17·98	29·1	**302**	3·02	1·738	9·120	27·54	47·8
263	2·63	1·621	6·917	18·19	29·5	**303**	3·03	1·740	9·181	27·82	48·5
264	2·64	1·624	6·970	18·39	29·9	**304**	3·04	1·743	9·241	28·09	48·9
265	2·65	1·627	7·022	18·61	30·3	**305**	3·05	1·746	9·302	28·37	49·5
266	2·66	1·630	7·075	18·82	30·7	**306**	3·06	1·749	9·363	28·65	50·1
267	2·67	1·634	7·129	19·03	31·1	**307**	3·07	1·752	9·425	28·93	50·7
268	2·68	1·637	7·182	19·25	31·5	**308**	3·08	1·755	9·486	29·22	51·3
269	2·69	1·640	7·236	19·46	31·9	**309**	3·09	1·757	9·548	29·50	51·8
270	2·70	1·643	7·290	19·68	32·3	**310**	3·10	1·760	9·610	29·79	52·4
271	2·71	1·646	7·344	19·90	32·75	**311**	3·11	1·763	9·672	30·08	53·0
272	2·72	1·649	7·398	20·12	33·2	**312**	3·12	1·766	9·734	30·37	53·6
273	2·73	1·652	7·452	20·34	33·6	**313**	3·13	1·769	9·796	30·66	54·2
274	2·74	1·655	7·507	20·57	34·0	**314**	3·14	1·772	9·859	30·96	54·9
275	2·75	1·658	7·562	20·79	34·5	**315**	3·15	1·775	9·922	31·25	55·4
276	2·76	1·661	7·617	21·02	34·9	**316**	3·16	1·777	9·985	31·55	56·1
277	2·77	1·664	7·673	21·25	35·3	**317**	3·17	1·780	10·04	31·85	56·6
278	2·78	1·667	7·728	21·48	35·8	**318**	3·18	1·783	10·11	32·15	57·3
279	2·79	1·670	7·784	21·71	36·2	**319**	3·19	1·786	10·17	32·46	58·0
280	2·80	1·673	7·840	21·95	36·7	**320**	3·20	1·788	10·24	32·76	58·5
281	2·81	1·676	7·896	22·19	37·2	**321**	3·21	1·791	10·30	33·07	59·2
282	2·82	1·679	7·952	22·42	37·6	**322**	3·22	1·794	10·36	33·38	59·8
283	2·83	1·682	8·008	22·66	38·08	**323**	3·23	1·797	10·43	33·70	60·5
284	2·84	1·685	8·065	22·90	38·6	**324**	3·24	1·80	10·49	34·01	61·2
285	2·85	1·688	8·122	23·15	39·0	**325**	3·25	1·803	10·56	34·33	61·9
286	2·86	1·691	8·179	23·39	39·5	**326**	3·26	1·805	10·62	34·64	62·6
287	2·87	1·694	8·236	23·64	40·0	**327**	3·27	1·808	10·69	34·96	63·2
288	2·88	1·697	8·294	23·88	40·5	**328**	3·28	1·811	10·76	35·28	63·9
289	2·89	1·70	8·352	24·13	41·0	**329**	3·29	1·814	10·82	35·61	64·6
290	2·90	1·703	8·410	24·39	41·5	**330**	3·30	1·816	10·89	35·93	65·2
291	2·91	1·705	8·468	24·64	42·0	**331**	3·31	1·819	10·95	36·26	65·9
292	2·92	1·709	8·526	24·89	42·5	**332**	3·32	1·822	11·02	36·59	66·7
293	2·93	1·711	8·585	25·15	43·0	**333**	3·33	1·825	11·09	36·92	67·3
294	2·94	1·714	8·643	25·41	43·5	**334**	3·34	1·827	11·15	37·26	68·1
295	2·95	1·717	8·702	25·67	44·0	**335**	3·35	1·830	11·22	37·59	68·8
296	2·96	1·720	8·761	25·93	44·6	**336**	3·36	1·833	11·29	37·93	69·5
297	2·97	1·723	8·820	26·19	45·1	**337**	3·37	1·835	11·35	38·27	70·2
298	2·98	1·726	8·880	26·46	45·6	**338**	3·38	1·838	11·42	38·61	71·0
299	2·99	1·729	8·940	26·73	46·2	**339**	3·39	1·841	11·49	38·96	71·7

TABLE XIII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON—(*continued*).

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3\cdot5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3\cdot5}$
340	3·40	1·844	11·56	39·30	72·5	**380**	3·80	1·949	14·44	54·87	106·9
341	3·41	1·846	11·63	39·65	73·2	**381**	3·81	1·952	14·51	55·30	108·0
342	3·42	1·849	11·69	40·00	74·0	**382**	3·82	1·954	14·59	55·74	109·0
343	3·43	1·852	11·76	40·35	74·8	**383**	3·83	1·957	14·66	56·18	110·0
344	3·44	1·854	11·83	40·70	75·5	**384**	3·84	1·959	14·74	56·62	111·0
345	3·45	1·857	11·90	41·06	76·3	**385**	3·85	1·962	14·82	57·06	112·0
346	3·46	1·860	11·97	41·42	77·0	**386**	3·86	1·964	14·90	57·51	113·0
347	3·47	1·862	12·04	41·78	77·8	**387**	3·87	1·967	14·97	57·96	114·0
348	3·48	1·865	12·11	42·14	78·5	**388**	3·88	1·969	15·05	58·41	115·0
349	3·49	1·868	12·18	42·51	79·4	**389**	3·89	1·972	15·13	58·86	116·0
350	3·50	1·871	12·25	42·87	80·1	**390**	3·90	1·975	15·21	59·32	117·1
351	3·51	1·873	12·32	43·24	81·0	**391**	3·91	1·977	15·29	59·77	118·1
352	3·52	1·876	12·39	43·61	81·8	**392**	3·92	1·980	15·36	60·23	119·2
353	3·53	1·879	12·46	43·98	82·5	**393**	3·93	1·982	15·44	60·70	120·4
354	3·54	1·881	12·53	44·36	83·5	**394**	3·94	1·985	15·52	61·16	121·5
355	3·55	1·884	12·60	44·74	84·3	**395**	3·95	1·987	15·60	61·63	122·6
356	3·56	1·887	12·67	45·12	85·2	**396**	3·96	1·990	15·68	62·10	123·6
357	3·57	1·889	12·74	45·50	85·9	**397**	3·97	1·992	15·76	62·57	124·7
358	3·58	1·892	12·81	45·88	86·8	**398**	3·98	1·995	15·84	63·04	125·8
359	3·59	1·895	12·89	46·27	87·6	**399**	3·99	1·997	15·92	63·52	126·9
360	3·60	1·897	12·96	46·65	88·5	**400**	4·00	2·00	16·00	64·00	128·0
361	3·61	1·90	13·03	47·04	89·5	**401**	4·01	2·002	16·08	64·48	129·0
362	3·62	1·902	13·10	47·44	90·3	**402**	4·02	2·005	16·16	64·96	130·0
363	3·63	1·905	13·17	47·83	91·1	**403**	4·03	2·007	16·24	65·45	131·1
364	3·64	1·908	13·25	48·22	92·0	**404**	4·04	2·009	16·32	65·94	132·3
365	3·65	1·910	13·32	48·62	92·9	**405**	4·05	2·012	16·40	66·43	133·6
366	3·66	1·913	13·39	49·02	93·9	**406**	4·06	2·014	16·48	66·92	134·9
367	3·67	1·915	13·47	49·43	94·7	**407**	4·07	2·017	16·56	67·42	136·0
368	3·68	1·918	13·54	49·83	95·5	**408**	4·08	2·019	16·64	67·91	137·0
369	3·69	1·920	13·61	50·24	96·5	**409**	4·09	2·022	16·72	68·41	138·2
370	3·70	1·923	13·69	50·65	97·4	**410**	4·10	2·024	16·81	68·92	139·4
371	3·71	1·926	13·76	51·06	98·3	**411**	4·11	2·027	16·89	69·42	140·5
372	3·72	1·928	13·84	51·48	99·3	**412**	4·12	2·029	16·97	69·93	141·6
373	3·73	1·931	13·91	51·89	100·0	**413**	4·13	2·032	17·05	70·44	143·0
374	3·74	1·934	13·99	52·31	101·0	**414**	4·14	2·034	17·13	70·95	144·1
375	3·75	1·936	14·06	52·73	102·0	**415**	4·15	2·037	17·22	71·47	145·4
376	3·76	1·939	14·13	53·15	103·1	**416**	4·16	2·039	17·30	71·99	146·6
377	3·77	1·941	14·21	53·58	104·1	**417**	4·17	2·042	17·39	72·51	148·0
378	3·78	1·944	14·29	54·01	105·0	**418**	4·18	2·044	17·47	73·03	149·1
379	3·79	1·946	14·36	54·44	105·9	**419**	4·19	2·046	17·55	73·56	150·3

TABLE XIII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON—(*continued*).

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$
420	4·20	2·049	17·64	74·08	151·5	**460**	4·60	2·144	21·16	97·33	208·6
421	4·21	2·051	17·72	74·62	153·0	**461**	4·61	2·147	21·25	97·97	210·0
422	4·22	2·054	17·80	75·15	154·2	**462**	4·62	2·149	21·34	98·61	211·8
423	4·23	2·056	17·89	75·68	155·4	**463**	4·63	2·151	21·43	99·25	213·2
424	4·24	2·059	17·97	76·22	156·9	**464**	4·64	2·154	21·53	99·89	215·1
425	4·25	2·061	18·06	76·76	158·1	**465**	4·65	2·156	21·62	100·5	217·0
426	4·26	2·064	18·14	77·31	159·5	**466**	4·66	2·158	21·71	101·2	218·2
427	4·27	2·066	18·23	77·85	160·6	**467**	4·67	2·161	21·81	101·8	220·0
428	4·28	2·068	18·31	78·40	162·0	**468**	4·68	2·163	21·90	102·5	221·7
429	4·29	2·071	18·40	78·95	163·2	**469**	4·69	2·165	21·99	103·1	223·1
430	4·30	2·073	18·49	79·50	164·8	**470**	4·70	2·168	22·09	103·8	225·0
431	4·31	2·076	18·57	80·06	166·3	**471**	4·71	2·170	22·18	104·4	226·7
432	4·32	2·078	18·66	80·62	167·5	**472**	4·72	2·172	22·27	105·1	228·0
433	4·33	2·080	18·74	81·18	168·8	**473**	4·73	2·175	22·37	105·8	230·1
434	4·34	2·083	18·83	81·74	170·3	**474**	4·74	2·177	22·46	106·4	231·8
435	4·35	2·085	18·92	82·31	171·5	**475**	4·75	2·179	22·56	107·1	233·1
436	4·36	2·088	19·00	82·88	173·0	**476**	4·76	2·181	22·65	107·8	235·0
437	4·37	2·090	19·09	83·45	174·3	**477**	4·77	2·184	22·75	108·5	237·0
438	4·38	2·092	19·18	84·02	175·7	**478**	4·78	2·186	22·85	109·2	238·6
439	4·39	2·095	19·27	84·60	177·2	**479**	4·79	2·188	22·94	109·9	240·1
440	4·40	2·097	19·36	85·18	178·4	**480**	4·80	2·190	23·04	110·6	242
441	4·41	2·100	19·45	85·76	180·0	**481**	4·81	2·193	23·13	111·3	244
442	4·42	2·102	19·53	86·35	181·5	**482**	4·82	2·195	23·23	112·0	246
443	4·43	2·104	19·62	86·94	182·8	**483**	4·83	2·197	23·33	112·6	247
444	4·44	2·107	19·71	87·53	184·2	**484**	4·84	2·200	23·42	113·3	249
445	4·45	2·109	19·80	88·12	185·7	**485**	4·85	2·202	23·52	114·1	251·5
446	4·46	2·111	19·89	88·71	187·1	**486**	4·86	2·204	23·62	114·8	253
447	4·47	2·114	19·98	89·31	188·5	**487**	4·87	2·206	23·71	115·5	255
448	4·48	2·116	20·07	89·91	190·0	**488**	4·88	2·209	23·81	116·2	256·5
449	4·49	2·119	20·16	90·51	191·6	**489**	4·89	2·211	23·91	116·9	258·2
450	4·50	2·121	20·25	91·12	193·1	**490**	4·90	2·213	24·01	117·6	260
451	4·51	2·123	20·34	91·73	194·8	**491**	4·91	2·216	24·11	118·3	262
452	4·52	2·126	20·43	92·34	196·2	**492**	4·92	2·218	24·20	119·0	264
453	4·53	2·128	20·52	92·95	197·6	**493**	4·93	2·220	24·30	119·8	266
454	4·54	2·130	20·61	93·57	199·1	**494**	4·94	2·222	24·40	120·5	268
455	4·55	2·133	20·70	94·19	200·9	**495**	4·95	2·225	24·50	121·3	270
456	4·56	2·135	20·79	94·81	202·7	**496**	4·96	2·227	24·60	122·0	272
457	4·57	2·137	20·88	95·44	203·9	**497**	4·97	2·229	24·70	122·7	274
458	4·58	2·140	20·97	96·07	205·2	**498**	4·98	2·231	24·80	123·5	275·6
459	4·59	2·142	21·06	96·70	207·0	**499**	4·99	2·233	24·90	124·2	277·5

TABLE XIII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON—(*continued*).

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$
500	5·00	2·236	25·00	125·0	279·5	**540**	5·40	2·324	29·16	157·4	366
501	5·01	2·238	25·10	125·7	281	**541**	5·41	2·326	29·27	158·3	368
502	5·02	2·240	25·20	126·5	283	**542**	5·42	2·328	29·37	159·2	370
503	5·03	2·242	25·30	127·2	285	**543**	5·43	2·330	29·48	160·1	373
504	5·04	2·245	25·40	128·0	287	**544**	5·44	2·332	29·59	161·0	375
505	5·05	2·247	25·50	128·7	289	**545**	5·45	2·334	29·70	161·8	377
506	5·06	2·249	25·60	129·5	291	**546**	5·46	2·336	29·81	162·7	380
507	5·07	2·251	25·70	130·3	293	**547**	5·47	2·339	29·92	163·6	382
508	5·08	2·254	25·80	131·1	295	**548**	5·48	2·341	30·03	164·5	385
509	5·09	2·256	25·91	131·8	297	**549**	5·49	2·343	30·14	165·4	387
510	5·10	2·258	26·01	132·6	299	**550**	5·50	2·345	30·25	166·3	390
511	5·11	2·260	26·11	133·4	301	**551**	5·51	2·347	30·36	167·3	392
512	5·12	2·263	26·21	134·2	304	**552**	5·52	2·349	30·47	168·2	395
513	5·13	2·265	26·31	135·0	306	**553**	5·53	2·351	30·58	169·1	397
514	5·14	2·267	26·42	135·8	308	**554**	5·54	2·353	30·69	170·0	400
515	5·15	2·269	26·52	136·6	310	**555**	5·55	2·356	30·80	170·9	402
516	5·16	2·271	26·62	137·4	312	**556**	5·56	2·358	30·91	171·8	405
517	5·17	2·274	26·73	138·2	314	**557**	5·57	2·360	31·02	172·8	408
518	5·18	2·276	26·83	139·0	316	**558**	5·58	2·362	31·13	173·7	411
519	5·19	2·278	26·93	139·8	318	**559**	5·59	2·364	31·24	174·6	413
520	5·20	2·280	27·04	140·6	320	**560**	5·60	2·366	31·36	175·6	415
521	5·21	2·282	27·14	141·4	323	**561**	5·61	2·368	31·47	176·5	418
522	5·22	2·284	27·25	142·2	325	**562**	5·62	2·370	31·58	177·5	421
523	5·23	2·287	27·35	143·0	327	**563**	5·63	2·373	31·69	178·4	423
524	5·24	2·289	27·45	143·8	329	**564**	5·64	2·375	31·80	179·4	426
525	5·25	2·291	27·56	144·7	331	**565**	5·65	2·377	31·92	180·3	429
526	5·26	2·293	27·66	145·5	334	**566**	5·66	2·379	32·03	181·3	431
527	5·27	2·295	27·77	146·3	336	**567**	5·67	2·381	32·14	182·2	434
528	5·28	2·298	27·88	147·2	338	**568**	5·68	2·383	32·26	183·2	436
529	5·29	2·300	27·98	148·0	340	**569**	5·69	2·385	32·37	184·2	439
530	5·30	2·302	28·09	148·8	342	**570**	5·70	2·387	32·49	185·2	442
531	5·31	2·304	28·19	149·7	345	**571**	5·71	2·389	32·60	186·1	445
532	5·32	2·306	28·30	150·5	347	**572**	5·72	2·391	32·71	187·1	447
533	5·33	2·308	28·41	151·4	349	**573**	5·73	2·393	32·83	188·1	450
534	5·34	2·310	28·51	152·2	352	**574**	5·74	2·396	32·94	189·1	453
535	5·35	2·313	28·62	153·1	354	**575**	5·75	2·398	33·06	190·1	456
536	5·36	2·315	28·73	154·0	356·6	**576**	5·76	2·40	33·17	191·1	459
537	5·37	2·317	28·83	154·8	358	**577**	5·77	2·402	33·29	192·1	461
538	5·38	2·319	28·94	155·7	361	**578**	5·78	2·404	33·40	193·1	464
539	5·39	2·321	29·05	156·5	363	**579**	5·79	2·406	33·52	194·1	467

TABLE XIII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON—(*continued*).

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3\cdot5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3\cdot5}$
580	5·80	2·408	33·64	195·1	470	**620**	6·20	2·490	38·44	238·4	593
581	5·81	2·410	33·75	196·1	473	**621**	6·21	2·492	38·56	239·4	596
582	5·82	2·412	33·87	197·1	475	**622**	6·22	2·494	38·68	240·6	599
583	5·83	2·414	33·98	198·1	479	**623**	6·23	2·496	38·81	241·8	603
584	5·84	2·416	34·10	199·1	481	**624**	6·24	2·498	38·93	242·9	606
585	5·85	2·418	34·22	200·2	484	**625**	6·25	2·50	39·06	244·1	610
586	5·86	2·420	34·34	201·2	486	**626**	6·26	2·502	39·18	245·3	614
587	5·87	2·422	34·45	202·2	489	**627**	6·27	2·504	39·31	246·5	616
588	5·88	2·425	34·57	203·2	492	**628**	6·28	2·506	39·43	247·6	620
589	5·89	2·427	34·69	204·3	495	**629**	6·29	2·508	39·56	248·8	624
590	5·90	2·429	34·81	205·3	498	**630**	6·30	2·510	39·69	250·0	627
591	5·91	2·431	34·92	206·4	501	**631**	6·31	2·512	39·81	251·2	630
592	5·92	2·433	35·04	207·4	505	**632**	6·32	2·514	39·94	252·4	635
593	5·93	2·435	35·16	208·5	508	**633**	6·33	2·516	40·06	253·6	638
594	5·94	2·437	35·28	209·5	510	**634**	6·34	2·518	40·19	254·8	641
595	5·95	2·439	35·40	210·6	513	**635**	6·35	2·520	40·32	256·0	645
596	5·96	2·441	35·52	211·7	516	**636**	6·36	2·521	40·45	257·2	648
597	5·97	2·443	35·64	212·7	519	**637**	6·37	2·523	40·57	258·4	652
598	5·98	2·445	35·76	213·8	522	**638**	6·38	2·525	40·70	259·7	655
599	5·99	2·447	35·88	214·9	525	**639**	6·39	2·527	40·83	260·9	659
600	6·00	2·449	36·00	216·0	529	**640**	6·40	2·529	40·96	262·1	662
601	6·01	2·451	36·12	217·0	532	**641**	6·41	2·531	41·08	263·3	666
602	6·02	2·453	36·24	218·1	535	**642**	6·42	2·533	41·21	264·6	670
603	6·03	2·455	36·36	219·2	538	**643**	6·43	2·535	41·34	265·8	674
604	6·04	2·457	36·48	220·3	540	**644**	6·44	2·537	41·47	267·0	677
605	6·05	2·459	36·60	221·4	544	**645**	6·45	2·539	41·60	268·3	681
606	6·06	2·461	36·72	222·5	547	**646**	6·46	2·541	41·73	269·6	685
607	6·07	2·463	36·84	223·6	550	**647**	6·47	2·543	41·86	270·8	688
608	6·08	2·466	36·96	224·7	553	**648**	6·48	2·545	41·99	272·1	692
609	6·09	2·468	37·08	225·8	556	**649**	6·49	2·547	42·12	273·3	696
610	6·10	2·470	37·21	226·9	560	**650**	6·50	2·549	42·25	274·6	700
611	6·11	2·471	37·33	228·1	563	**651**	6·51	2·551	42·38	275·9	704
612	6·12	2·473	37·45	229·2	567	**652**	6·52	2·553	42·51	277·1	707
613	6·13	2·476	37·57	230·3	570	**653**	6·53	2·555	42·64	278·4	711
614	6·14	2·478	37·69	231·4	573	**654**	6·54	2·557	42·77	279·7	715
615	6·15	2·480	37·82	232·6	576	**655**	6·55	2·559	42·90	281·0	719
616	6·16	2·482	37·94	233·7	580	**656**	6·56	2·561	43·03	282·3	723
617	6·17	2·484	38·06	234·9	583	**657**	6·57	2·563	43·16	283·5	726
618	6·18	2·486	38·19	236·0	586	**658**	6·58	2·565	43·29	284·9	730
619	6·19	2·488	38·31	237·1	590	**659**	6·59	2·567	43·42	286·2	734

TABLE XIII.—MULTIPLIERS USED IN APPLYING THE LAW OF COMPARISON—(*continued*).

Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$	Ship Lgth.	l	$\sqrt{l}$	l^2	l^3	$l^{3.5}$
660	6·60	2·569	43·56	287·5	738	**685**	6·85	2·617	46·92	321·4	841
661	6·61	2·571	43·69	288·8	741	**686**	6·86	2·619	47·06	322·8	845
662	6·62	2·573	43·82	290·1	746	**687**	6·87	2·621	47·19	324·2	849
663	6·63	2·574	43·95	291·4	750	**688**	6·88	2·623	47·33	325·6	853
664	6·64	2·576	44·09	292·7	754	**689**	6·89	2·625	47·47	327·0	858
665	6·65	2·578	44·22	294·0	757	**690**	6·90	2·626	47·61	328·5	862
666	6·66	2·580	44·35	295·4	761	**691**	6·91	2·628	47·74	329·9	866
667	6·67	2·582	44·48	296·7	766	**692**	6·92	2·630	47·88	331·3	871
668	6·68	2·584	44·62	298·0	770	**693**	6·93	2·632	48·02	332·8	875
669	6·69	2·586	44·75	299·4	774	**694**	6·94	2·634	48·16	334·2	880
670	6·70	2·588	44·89	300·7	778	**695**	6·95	2·636	48·30	335·7	885
671	6·71	2·590	45·02	302·1	782	**696**	6·96	2·638	48·44	337·1	890
672	6·72	2·592	45·15	303·4	786	**697**	6·97	2·640	48·58	338·6	894
673	6·73	2·594	45·29	304·8	790	**698**	6·98	2·642	48·72	340·0	898
674	6·74	2·596	45·42	306·1	795	**699**	6·99	2·644	48·86	341·5	903
675	6·75	2·598	45·56	307·5	799	**700**	7·00	2·645	49·00	343·0	907
676	6·76	2·600	45·69	308·9	803	**705**	7·05	..	..	350·4	931
677	6·77	2·601	45·83	310·2	807	**710**	7·10	..	..	357·9	953
678	6·78	2·603	45·96	311·6	811	**715**	7·15	..	..	365·5	978
679	6·79	2·605	46·10	313·0	815	**720**	7·20	..	..	373·2	1002
680	6·80	2·607	46·24	314·4	819	**725**	7·25	..	..	381·0	1026
681	6·81	2·609	46·37	315·8	823	**730**	7·30	..	..	389·0	1050
682	6·82	2·611	46·51	317·2	828	**735**	7·35	..	..	397·0	1076
683	6·83	2·613	46·64	318·6	833	**760**	7·60	..	..	438·97	1208
684	6·84	2·615	46·78	320·0	837						

EXPERIMENT TANKS.

"Ship-model Experiment Tanks: their purpose and application." Paper by Prof. W. S. Abell, read before the Liverpool Engineering Society, 16th November 1910.

"Methodical Experiments with Mercantile Ship Forms." Paper by Mr G. S. Baker, read before the Institution of Naval Architects, 14th March 1913. (Discussion.)

"The National Experimental Tank and its Equipment." Paper by G. S. Baker, Esq., read before the Institution of Naval Architects, 5th April 1911.

TABLE XIV.—SOME EXPERIMENT TANKS.*

No.	Date of beginning experimental work.	Proprietor.	Place.	Dimensions in feet. Length.	Breadth.	Depth.	Run.	Area of cross-section in sq. ft.	Maximum velocity in feet per second.
1	1884	W. Denny & Bros.	Dumbarton, Scotland.	300	22	10	250		
2	1886	British Admiralty.	Haslar, England.	400	20	9	360	170	16
3	1889	Italian Government.	Spezia, Italy.	538	19·7	9·9	480		
4	1892	"Kette" S.B. Co. (old Uebigau tank). Now discontinued.	Uebigau, near Dresden, Germany.	206	24·6	4·5	206		
5	1893	Russian Government.	St Petersburg.	441	21·8	11	374		
6	1899	U.S. Government.	Washington, U.S.A.	470	42·7	14·7	384	418	30
7	1900	Cornell University.	Ithaca, N.Y., U.S.A.	418	16	10	418		
8	1900	N. German Lloyd S.S. Co.	Bremerhaven.	541	19·7	10·5	476		
9	1902	Tech. Hochschule and German Government.	Berlin.	557·7	34·4	11·5	479	265	23
10	1903	John Brown & Co.	Clydebank, Scotland.	445	20	9	400	180	16
11	1904	Tech. Hochschule and Saxon Govt. (new Uebigau tank).	Uebigau, near Dresden, Germany.	312	21·0	11·3	288	200	16
12	1906	University of Michigan.	Ann Arbor, Michigan, U.S.A.	300	22	10	275	200	13
13	1906	French Government.	Paris.	525	32·8	13·1	442	326	15
14	1907	Mitsubishi S.B. Co.	Nagasaki, Japan.	..	20	12	450	235	20
15	1911	National Physical Laboratory.	Teddington, Herts, England.	550	30	12·25	494	360	25
16	1911	,, ,, ,,	Do. (small tank).	68	5	3·25			
17	..	....	Hamburg.	..	26·2	21·0	1083	..	36 (estd.)
18	..	Vickers Ltd.	Barrow, England.	..	20	..	420		
19	..	....	Vienna.	..	32·8	16·4	550	..	24

The length and breadth are over all at the water surface; the depth is at the centre line.

* Some of these particulars were obtained from Mr H. A. Everett's illustrated article on the subject, in *International Marine Engineering*, January 1909, and some from Mr G. S. Baker's book, *Ship Form, Resistance and Screw Propulsion*.

Large and Small Experimental Tanks.

An excellent article on this subject appeared in *The Engineer*, of 3rd May 1912. Some letters in the *Journal of Commerce* about October 1910 pointed out disadvantages of small tanks on similar grounds. With any small tank there are inevitable inaccuracies, but it may be useful for preliminary weeding out of unsuitable models.

In the Caws tank at Sunderland the models are suspended pendulum fashion and swung through the water, the resistance being measured at the position between the first half of its swing when its speed is accelerating, and the second half when it is decelerating. At the vertex of the swing the wave system cannot be considered developed in a manner proper to the instantaneous speed of the model. In Herr Wellenkamp's tank the model was towed by a falling weight, but there was a difficulty in keeping the model straight on its course; and there were other difficulties common to all small tanks with small-sized models, such as inertia, relatively large differences in friction of the towing gear, capillarity, and surface tension.

With large tanks on the Froude system, which has stood the test of forty-five years, the models are run at a steady speed; the measuring gear records an exact measurement of the resistance for the whole length of the run, and every result so obtained, by the expert in charge of the tank, is solid groundwork upon which unending analyses and estimates can always be based.

Models are frequently made about 15 ft. long, and are usually of paraffin wax. The models at the United States Model Basin are 20 ft. in length, and are made of wood. For the "Manning" the length of the model was $23\frac{1}{2}$ ft. As the size of the model is increased, the magnification of results and the probable error are decreased. With very small models the forces measured would be very small, and would be liable to excessive error. Suppose that for a 450-ft. ship we had a 15-ft. model, the resistance of the ship would be $(30)^3$ or 27 000 times that of the model (since $\frac{450}{15} = 30$). If the model were 10 ft. long, the relative resistances would be as 1 : 90 000. The resistances recorded would be from about ·5 lb. upwards for 15-ft. models. Mr D. W. Taylor's table ix, showing results of tank model experiments for the "Yorktown," give resistances for his 20-ft. models of from 1·1 lb. to 93 lbs.

The effect of temperature of the tank water upon resistance was noted in the discussion on Mr Baker's paper in 1913.

Sir Archibald Denny mentioned that Mr Mumford had said that it had been well established that a difference of 5 per cent. in resistance was caused by a difference of 12° Fahr., the resistance increasing with the fall of temperature; also that, probably due to a changing difference in temperature between one end and the other end of the tank, there was an absolute movement in the water—in one direction in summer, and in the other direction in winter. At the Bushey tank Mr Baker employed a float half-way down the useful length of the tank, and the movements of the float are noted.

The records of the work done by Mr R. E. Froude and Mr G. S. Baker afford a splendid illustration of the value of experimental tank research work. Whenever an appreciable departure, from forms already in commission, is proposed, models should be made and tested individually. In practice, in the preliminary design stage, the problem is to determine the dimensions and form most suitable to specified conditions, not only from the point of view of resistance, but from that of the fulfilment of conditions such as draught and stability, trim, machinery space, capacity for cargo, sea performance, etc.

The procedure is for the shipowner to give the National Physical Laboratory, or other experiment tank works, a copy of the lines of an existing type-ship, from which the superintendent of the tank makes a model, the shipowner stating the limits of variation of the load water-line, to provide sufficient stability, and the limits within which the shape of the curve of sectional areas may be allowed to vary to suit the arrangement of machinery, etc. The experiment tank authority then conducts a set of trials of the first model in the tank, and offers other models having different positions of the longitudinal centre of buoyancy, suggesting a model perhaps better than the parent form from the point of view of propulsion. The shipowner finally selects the one which he considers the best obtainable for speed consistent with other requirements of the service.*

The results of the first years of Mr G. S. Baker's testing of merchant ship models at the Froude tank almost invariably showed that the cost of the test was saved on the fuel bill of the ship in the first six months of its running. In his experiments with models of ships building or contemplated, Mr Baker and his staff have been successful in effecting reductions in the power to the extent, in some cases, of as much as 25 per cent. There is no doubt that, as Professor W. S. Abell has remarked when recommending the use of experimental tanks, "the economical

* Tank trials give resistance and, its equivalent, E.H.P.

performances of mercantile vessels of moderate speeds could be considerably improved if proper investigation of form, propeller, and the combination of the propeller and ship were made."

In addition to the commercial side of the work at the tank, much valuable research of a general nature has been carried out on ship forms. There are, however, large unexplored fields for methodical experiment, not only to fill the gaps between the fine and full types already dealt with in England and America, but also to treat full models of the cargo type, of broader and shorter proportions than have hitherto been exhaustively tested.

Still, from the data already published on types ranging from Mr Froude's fine-lined warships and Mr Taylor's moderately fine vessels to the merchant ship forms tested by Professor Sadler and Mr Baker, shipowners and shipbuilders may, without having models of their own, predict with some degree of accuracy, for many types of ship, the power required at a given speed. One object of this book is to put a collection of such published results in a form easily accessible for reference, and to illustrate methods of putting these results to practical use.

The reader is referred to the original papers by the authorities quoted; our intention is rather to present quantitative results, to give figures to multiply by in the everyday problem of settling powers and speeds, and to attempt to compare figures obtained from tank trials with the power figures deduced from service performances of actual ships. Though it is universally agreed that in tank trials the results are obtained with even greater accuracy than in full-sized trials, that differences of resistances developed at different draughts and trims are in the same direction as those with the actual vessel, and that there is a great resemblance in character between the "curves of resistance" of the model and of the ship—the humps and hollows occurring at similar speeds,—it is also true that results from even the best tank experiments may be misleading when used for obtaining actual values; but that is no reason for ignoring them.

To quote from *The Engineer*: "It should never be forgotten that the builder who adopts the experimental method has not only the same information at his disposal from his trials on the measured mile as one who has no tank, but he has his model results in addition, and it is in co-ordination of these that the strength of his position lies." Tank trials made with models of existing ships, especially those for which the records of progressive trials are available, are particularly instructive, and provide the best means of arriving at the propulsive efficiencies or ratios of effective horse-power to indicated horse-power, shaft

horse-power, or brake horse-power. In other words, the tank test is not entirely complete until the ship trial is made. The determination of the propulsive efficiency completes the experiment. These "back steamers" are always valuable for reference for enabling us to predict the speed of any given steamer attainable by a given I.H.P., S.H.P., or B.H.P. On Plates 30–2, 35 will be found curves of this ratio $\frac{\text{E.H.P.}}{\text{I.H.P.}}$, or propulsive efficiency, or propulsive coefficient, as it is sometimes called. By keeping results of model experiments in touch with those of the completed ship, the correct percentage additions to allow in design may be determined, as between tank trial and measured mile trial, or between tank trial and performance on voyage. Unfortunately progressive trials are very rare, and when they are run the draught of ship is too light in many cases.

By means of a properly arranged dynamometer, when towing a ship or model through still water, we can measure the net or tow-rope resistance, or total resistance, which is made up of four components, viz.: frictional, wave-making, eddy-making, and air resistance. The model is usually run "naked," *i.e.* without appendages, such as bilge keels, bossings, shafts, rudder, etc.; these, of course, should be added when computing the wetted surface of the actual ship, and their effect on the eddy-making resistance, and the hull-appendage factor, taken into account.*

A sure method of determining the resistance of a ship is to tow her through still water, from a long outrigged boom, at various speeds, and note the resistances, as was done in the case of the "Greyhound," where special devices were fitted in order that only the horizontal component of the force on the tow-rope was measured (*Trans. Inst. Naval Architects*, 1874, Froude); but it is seldom that experiments are carried out on such a large scale. In the *Transactions of the American Society of Naval Architects and Marine Engineers*, 1911, Professor C. H. Peabody gave the results of towing the "Froude," a miniature steamer 37·6 ft. in length. In the case of the 760-ft. Cunard liner "Mauretania," the builders made exhaustive propeller and other experiments with an exactly similar vessel 37 ft. in length.

Estimating Horse-Power from Model Experiments.

Take the case of a model of a twin-screw steamer :—418 ft. b.p. × 52 ft. beam × 23 ft. mean draught, 9 100 tons displacement.

* See paper by Commander Dyson, U.S.N., read before the American Society of Naval Engineers, *Transactions*, 22.

Paraffin model, 14 ft. long, towed at various speeds; tow-rope resistance $r = 2{\cdot}6$ lbs. at the speed corresponding to $14\frac{1}{2}$ knots of the full-sized ship.

(1) *Speed*:—

Let this speed of model be V_m.

Then
$$\frac{V_m}{14\frac{1}{2}\text{ knots}} = \frac{\sqrt{\text{length of model}}}{\sqrt{\text{length of ship}}} = \frac{\sqrt{14}}{\sqrt{418}}.$$

The various square roots, squares, cubes, etc., may be conveniently taken from Table XIII, pp. 38–46, as multipliers or functions of l.

$$\therefore\ V_m = 2{\cdot}66 \text{ knots.}$$

(2) *Wetted surface*:—

Let S_m = wetted surface of model = 34 sq. ft.

and S = wetted surface of ship = 30 300 sq. ft.

$$\frac{S_m}{S} = \frac{l_1^2}{l^2} = \frac{{\cdot}019\,6}{17{\cdot}47}.$$

The square of l being taken from Table XIII as before.

(3) *Skin frictional resistance*, r_f:—f (for model) = ·008 83. $n = 1{\cdot}94$. (From Table I, Tideman's Fresh-Water Constants.)

$$\begin{aligned} r_f &= f \times \text{wetted surface} \times (V_m)^{1{\cdot}94} \\ &= {\cdot}008\,83 \times 34 \times (2{\cdot}66)^{1{\cdot}94} \\ &= {\cdot}008\,83 \times 34 \times 6{\cdot}662 = 2 \text{ lbs.} \end{aligned}$$

(4) *Residuary Resistance of model*:—

$$\begin{aligned} r_r &= r - r_f \\ &= 2{\cdot}6 - 2{\cdot}0 \\ &= {\cdot}6 \text{ lb.} \end{aligned}$$

r = total resistance.

r_r = residuary resistance.

(5) *The corresponding Residuary Resistance of the full-sized ship*, R_w, follows from the Law of Comparison, thus:—

$$\begin{aligned} \frac{R_w}{r_r} &= \frac{36}{35}\left(\frac{L^3}{l^3}\right) = \frac{36}{35}\left(\frac{l^3}{l_1^3}\right) \\ &= \frac{36}{35} \times \frac{73{\cdot}03}{{\cdot}002\,744} = 27\,400. \end{aligned}$$

$$\therefore\ R_w = r_r \times 27\,400 = 16\,450 \text{ lbs.}$$

The ratio $\frac{36}{35}$ is used when passing from fresh water to salt water.

(6) *The Residuary H.P. for the full-sized ship*:—

$$\begin{aligned} &= {\cdot}003\,07 \times R_w \times V \\ &= {\cdot}003\,07 \times 16\,450 \times 14{\cdot}5 \\ &= 733. \end{aligned}$$

(7) *Skin H.P. of full-sized ship in salt water:—*
Take Froude's table for salt water. $f = \cdot 008\ 85$ from Table V.

$$\begin{aligned} n &= 1\cdot 83 \text{ for resistance and } 2\cdot 83 \text{ for power} \\ \text{Skin H.P.} &= f \times \text{wetted surface} \times \cdot 003\ 07 \times (V)^{2\cdot 83} \\ &= \cdot 008\ 85 \times 30\ 300 \times \cdot 003\ 07 \times 1\ 935 \\ &= 1\ 592. \end{aligned}$$

(8) *The total E.H.P. for the full-sized ship in salt water at* $14\frac{1}{2}$ *knots:—*

$$\begin{aligned} \text{Total E.H.P.} &= \text{Skin H.P.} + \text{Residuary H.P.} \\ &= 1\ 592 + 733. \\ \therefore \text{Total E.H.P.} &= 2\ 325. \end{aligned}$$

This is the E.H.P. from the naked model.

If the I.H.P. is 4 650 at $14\frac{1}{2}$ knots, the propulsive coefficient

$$\frac{\text{E.H.P. (naked)}}{\text{I.H.P.}} = \cdot 50.$$

With Taylor's skin friction constants for model, $f =$ about $\cdot 010\ 03$ and $n = 1\cdot 854$, the skin frictional resistance of the model would have been about the same.

$$\begin{aligned} r_f &= \cdot 010\ 03 \times 34 \times (2\cdot 66)^{1\cdot 854} \\ &= \cdot 010\ 03 \times 34 \times 5\cdot 82 \\ &= 2\cdot 0 \text{ lbs.} \end{aligned}$$

Then

$$r_r = \cdot 60 \text{ lb. as before}$$

and

$$R_w = 16\ 450 \text{ lbs.}$$

Residuary H.P. for ship would have been

$$\begin{aligned} &= \cdot 003\ 07 \times 16\ 450 \times 14\cdot 5 \\ &= 733 \text{ as before.} \end{aligned}$$

Skin H.P. for ship, if Tideman's skin frictional constants had been taken, would have been

$$\begin{aligned} &= \cdot 009\ 085\ 6 \times 30\ 300 \times \cdot 003\ 07 \times 1\ 935 \\ &= 1\ 635 \\ \text{Total E.H.P.} &= 1\ 635 + 733 = 2\ 368 \\ \frac{\text{E.H.P. (naked)}}{\text{I.H.P.}} &= \cdot 51. \end{aligned}$$

It does not matter much whether we take Tideman's fresh-water figures, $n = 1\cdot 94$, or Taylor's fresh water $n = 1\cdot 854$, and the constants used at the U.S. tank, so far as the model is concerned. For the skin H.P. of the full-sized ship, in design

work, perhaps it is better to use Tideman's salt-water constants (subject to $n\sqrt{\ } = 1{\cdot}83$).

For investigating figures by Mr R. E. Froude, Mr Luke, and Mr Baker, Froude's skin constants from the O and (c) values should be taken, as they usually give lower skin friction power and higher residuary H.P. The differences, however, are slight.

Our tables of skin H.P. per 1 000 sq. ft. of wetted surface provide an easy means of reckoning the skin H.P.; for instance, in the above example,

$$\text{Skin H.P.} = 54{\cdot}33 \times 30{\cdot}3 = 1\,645.$$

Displacement of 14-ft. model of 418-ft. ship.

Model, $14 \times 1{\cdot}742 \times {\cdot}771$. Block coefficient $= {\cdot}637$. Displacement $= {\cdot}334$ ton in fresh water $= \dfrac{14 \times 1{\cdot}742 \times {\cdot}771 \times {\cdot}637}{36} = {\cdot}334$ ton.

Another way to calculate the displacement of the model is to take the ratio of the cubes of the lengths of the model and the ship, and multiply this ratio by the displacement of the full-sized ship and by $\frac{35}{36}$ in passing from salt water to fresh water; thus

$$\frac{35}{36} \times \frac{{\cdot}002\,744}{73{\cdot}03} \times 9\,100 = {\cdot}334 \text{ ton in fresh water.}$$

The displacement of the model is 747 lbs. in fresh water (δ).

The residuary resistance of model in lbs. per ton of displacement $= \dfrac{{\cdot}6}{{\cdot}334} = 1{\cdot}8$.

The corresponding residuary resistance of the ship (after calculating skin friction separately) in lbs. per ton of displacement $= \dfrac{16\,450}{9\,100} = 1{\cdot}8$.

This is based upon ·6 lb. residuary resistance of model.

If we gave the skin frictional resistance the 10 per cent. addition for form, and based our residuary resistance upon ·4 lb. for model, the residuary resistance per ton of displacement for ship or model would be 1·2 lb., and this agrees with Taylor's contours. Taylor, however, so far as we know, did not make the allowance for added skin friction due to form.

Total Resistance of Ship Model, 14 ft. long in fresh water, representing a twin-screw steamer $418 \times 52 \times 23$ ft. mean draught, 9 100 tons displacement, $14\frac{1}{2}$ knots speed.

Let V_m = the corresponding speed of the model = 2·66 knots, and its resistance at that speed 2·6 lbs.

	Lbs. resistance.	H.P.
I. *Skin frictional resistance.*		
(1) The hull proper, or naked hull, including an ordinary amount of deadwood. $= f \times$ wetted surface in sq. ft. $\times (V_m)^{1\cdot 94}$ $= \cdot 008\,83 \times 34 \times (2\cdot 66)^{1\cdot 94}$ $= \cdot 008\,83 \times 34 \times 6\cdot 662 = 2$ lbs. . .	2·00	
(2) f will vary with temperature of tank water. At the National Physical Laboratory, where paraffin models are used, Mr Baker deducts 3 per cent. from the calculated skin frictional resistance for an increase of 10 degrees Fahr. temperature of water. (Plus or minus according to temperature)	+ or −	
(3) The surface of appendages, such as bilge keels, propeller struts, shaft bossings, rudder, and deadwood in excess of the ordinary amount, if there are any appendages on the model when it is tested, is calculated and added to the wetted surface of the naked hull. (Models are almost always tested naked, *i.e.* without the appendages.)		
(4) A percentage addition to the calculated skin frictional resistance (given as 5 per cent. to 20 per cent. by Mr Baker), depending upon fulness of form. Over and above the skin frictional resistance calculated from Mr W. Froude's and Tideman's values of f for planes, there is an excess resistance accounted for by the increase in mean rubbing velocity between the streams and the ship form. Say 10 per cent. in this case . .	0·20	
II. *Eddy-making resistance.* A small item with a naked model . .	(Almost negligible)	
III. *Wave-making Resistance.* The sum of the eddy-making and wave-making = total resistance − skin frictional resistance.		

	Lbs. resistance.	H.P.
This assumes that we neglect air resistance, which in the case of a tank experiment is such a minute quantity that it may well be left out of account. $2\cdot6 - 2\cdot2 = \cdot40$ lb.	·40	
IV. *Total water resistance* = I + II + III = 2·60 lbs.	2·60	

Total E.H.P. of Full-sized Ship, deduced from the foregoing model results. Passenger ship, twin-screw, 418 × 52 × 23 ft. mean draught, 9 100 tons displacement, 14½ knots speed. Wetted surface from Mumford's formula = 30 300 sq. ft.

	Lbs. resistance.	H.P.
I. *Skin friction.*		
Skin frictional resistance, R_f.		
(1) The hull proper, or naked hull, including an ordinary amount of deadwood $= f \times$ wetted surface in sq. ft. $\times V^{1\cdot83}$ $f = \cdot008\,85$ from Froude's figures. $n = 1\cdot83$ $R_f = \cdot008\,85 \times 30\,300 \times (14\cdot5)^{1\cdot83}$ $= \cdot008\,85 \times 30\,300 \times 133\cdot4 = 35\,800$ lbs.	35 800	...
(2) *Skin H.P.* = ·003 070 7 × skin resistance in lbs × V = 1 592.		
(3) *The Skin H.P.* is usually calculated without first reckoning the skin frictional resistance, thus:— *Skin H.P.* $= f \times$ wetted surface $\times \cdot003\,070\,7 \times V^{2\cdot83}$ = 1592	...	1 592
(4) f will vary with temperature, but as most vessels pass from hot to cold climates, average values of f are taken.		
(5) The surface of appendages, such as bilge keels, propeller struts, shaft bossings, shafts, rudder, and deadwood in excess of the ordinary amount, is calculated, and added to the wetted surface of the naked hull.		

	Lbs. resistance.	H.P.
A better way, given by Mr Baker, is to calculate the rudder area separately, taking frictional coefficient for its own length, and velocity = (velocity of ship) (1 + slip ratio) $(1 - w)$. The bilge keels, if properly placed, are taken as additional wetted surface of the ship. The wetted surface of the shaft bossings may be added to the wetted surface of the ship. Total 6·07 per cent.	2 180	97
(6) A percentage addition to the calculated skin frictional power (given as 5 per cent. to 20 per cent. by Mr Baker, over and above Froude's plank value of f), depending for its amount upon fulness of form. Let us take 10 per cent. in this case, 3 580 lbs.	3 580	159·2
If Tideman's skin frictional constants were used instead of Froude's, the skin H.P. would be about 4½ per cent. in excess of Froude's skin H.P., and Mr Baker's percentage addition would have to be reduced by the 4½ per cent.		
II. (7) *Eddy-making*, due to irregular motion of rudder, water round propeller struts or shaft bossings, broken water around the stern-post, stem, bilge keels, and other appendages. The percentage for shaft bossings may be taken from Mr Baker's information, p. 378, say 3 per cent. For the eddying round other appendages about 1 per cent. may be added. Total, 4 per cent. of the wave-making resistance	365	16·3
III. *Wave-making*. The sum of the eddy-making, the wave-making, and the air resistances = residuary resistance = total resistance − skin frictional resistance.		
The wave-making resistance of ship is deduced from the wave-making resistance of model by the Law of Comparison. If r_w = ·4 lb. for the model, and R_w the wave-making resistance of the ship, L = length of ship, 418 ft. l = length of model = 14 ft.		

	Lbs. resistance.	H.P.
$\frac{R_w}{r_w} = \frac{36}{35}\left(\frac{L}{l}\right)^3$. ∴ $R_w = 10\,950$ lbs.	10 950	
Wave H.P. = ·003 070 7 × wave-making resistance × V = ·003 070 7 × 10 950 × 14·5 = 489	...	439
An addition may be required to allow for rolling and pitching, rough water tending to disturb the regular formation of waves and placing the ship in positions which cause the total average resistance to be increased. In a large ship these retardations are less than in the case of a small ship.	...	
IV. *Air Resistance.*		
Let A = the 'thwartship area in square feet of the above-water portion of the ship, moving normally to the direction of motion of the vessel, at a speed V in knots, and K = a constant, given by Rear-Admiral Taylor as ·003 5 to ·005. Then the air resistance in lbs., R, = K . A . V^2.		
In the case of our 418-ft. passenger liner, let A = 2 646 sq. ft.		
The horse-power absorbed in overcoming R is $\frac{R \times V \times 101\cdot33}{33\,000}$.		
V depends upon the fore and aft component of the relative velocities of the ship and the wind. If speed of ship = 14·5 knots against a 20-knot wind, then V = 34·5.		
Here R = ·004 3 × 2 646 × $(34\cdot5)^2$ = 13 500 lbs.	13 500	
Air H.P. (effective) = ·003 070 7 × 13 500 × 14·5 = 601	...	601
(The air resistance is taken at the speed of ship through the *water*.)		
Suppose propulsive coefficient to be ·47, then I.H.P. = $\frac{\text{E.H.P.}}{\text{Propulsive coefficient}} = \frac{2\,325}{\cdot47}$ = 4 950,		

	Lbs. resistance.	H.P.
and air I.H.P. $= \dfrac{601}{\cdot 47} = 1\,280$ against 20-knot wind.		
In calm air (no wind) $V = 14{\cdot}5.$ $(14{\cdot}5)^2 = 210.$		
$R = {\cdot}004\,3 \times 2\,646 \times 210 = 2\,390$ lbs.	2 390	
Air H.P. (effective) $= {\cdot}003\,070\,7 \times 2\,390 \times 14{\cdot}5 = 106.$		106
Air I.H.P. $= \dfrac{106}{\cdot 47} = 226.$		
$1\,280 - 226 = 1\,054$ I.H.P. difference.		
If $\dfrac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = 264 = \dfrac{(9\,100)^{\frac{2}{3}} \times (14{\cdot}5)^3}{4\,950}$ at 14·5 knots in calm air, then perhaps we may say that, approximately, there would be 1 054 I.H.P. less available for propelling the ship through the water when going against a 20-knot wind. Thus $4\,950 - 1\,054 = 3\,896.$		
If $\dfrac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = 264$, then $\dfrac{(9\,100)^{\frac{2}{3}} \times (13{\cdot}3)^3}{3\,896} = 264.$		
The speed of the ship against the 20-knot wind would be 13·3 knots, at the same gross I.H.P., viz. 4 950, which was required for 14½ knots in calm air.		
Or, if we took the gross I.H.P. in the usual way, $\dfrac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = \dfrac{(9\,100)^{\frac{2}{3}} \times (13{\cdot}3)^3}{4\,950} = 207.$		
V. Summing the figures which we have arrived at in our process of building up the power, we have:—		
Resistance:—		
Skin resistance $= 35\,800 + 2\,180 + 3\,580 = 41\,560$		
Eddy-making $= 35\,800 + 2\,180 + 3\,580 = 365$		
Wave-making $= 35\,800 + 2\,180 + 3\,580 = 10\,950$		
Calm air resistance $= 35\,800 + 2\,180 + 3\,580 = 2\,390$		
Total $= 55\,265$	55 265	

	Lbs. resistance.	H.P.
E.H.P. :—		
Skin H.P. = 1 592 + 97 + 159·2 = 1 848·2		
Eddy-making H.P. = 1 592 + 97 + 159·2 = 16·3		
Wave-making H.P. = 1 592 + 97 + 159·2 = 489		
Calm air H.P. = 1 592 + 97 + 159·2 = 106		
Total = 2 459·5	...	2 459·5
E.H.P. (naked) from model = 2 325.		
Gross E.H.P. (built up) = 2 459·3.		
i.e. appendages and air make a difference of 5¾ per cent. in calm air.		
Again, E.H.P. (naked) from model = 2 325.		
Gross E.H.P. (built up) = 2 954·5.		
1 848·2		
16·3		
489		
601		
2 954·5,		
or 26¾ per cent. addition for appendages and air when steaming against 20-knot wind.		
The average would be		
1 848·2 . . . Skin H.P.		
16·3 . . . Eddy ,,		
489 . . . Wave ,,		
353·5 . . . Wind ,,		
2 707·0 gross E.H.P.	...	2 707
which means an allowance of 16 per cent. for appendages and air, on the average weather, for the run out and home; and probably a better result might be expected, because on the outward run the wind might be a following one assisting the ship.		

This vessel suffered a reduction of a knot of speed at full power when steaming against a 20-knot wind, about 8 per cent. of the I.H.P. being absorbed in overcoming wind resistance.

CHAPTER IV.

CORRECTION FOR SKIN FRICTION.

GIVEN the dimensions of a ship, with displacement and other particulars.

From this we may *derive* any number of "similar ships." The linear dimensions of the *derived ship* are all directly proportional to the linear dimensions of the known vessel. The displacement of the *derived ship* and the displacement of the original vessel bear the same ratio to one another as the cubes of the linear dimensions. The speed of the *derived ship* is to the speed of the first vessel as the square root of the length of the derived ship is to the square root of the length of the first known ship. In other words, the displacement varies as $(\text{length})^3$; the speed varies as $\sqrt{\text{length}}$; and the horse-power to overcome the residuary resistance varies as $(\text{length})^{3 \cdot 5}$.

For comparing a model 14 ft. long, made of paraffin, and tried in a fresh-water tank, with a similar vessel 400 ft. long of clean painted steel for service in the salt sea, we use Tables I to VII, and other tables or curves made from them. Not only is the water of different density in the two cases, but the surfaces in contact with the water have, from their nature, different resistances to motion from other causes. For instance, the power of the speed at which the resistance is varying, or index (n) of variation of resistance with speed, is different in the two cases; the coefficients of fluid friction for the different lengths of surfaces are different from each also—all causing

$$\underset{\text{(for the model)}}{f \,.\, S \,.\, V^n} \text{ to be different from } \underset{\text{(for the ship)}}{f \,.\, S \,.\, V_n}.$$

The difference between the two is the amount of the skin friction correction.

Though no friction experiments on a large scale have been made, values of the coefficient of fluid friction for painted surfaces

up to 500 and 600 ft. long are included in tables based upon Froude's experiments with flat boards up to 50 ft. in length. The classical account of the experiments with H.M.S. "Greyhound," copper-sheathed gunboat (*Trans. Inst. Naval Arch.*, 1874, Froude), gave proof of the accuracy of the scale, which is now in constant use at experimental tank works in Great Britain, the Continent of Europe, and America. Other values of f, ascribed to Tideman, for clean painted ships in salt water, similar to Froude's constants, but about 5 per cent. higher, are given in Table I and used throughout this work for calculating skin friction horse-power and resistance of ships in salt water.

Table II gives values of the coefficient of skin friction for models in fresh water from Froude's figures, and Plate 1 gives Froude's values of f, with the corresponding values of n for various qualities of surface in fresh water.

Tables VIII and IX of skin frictional resistance and horse-power per 1 000 sq. ft. of wetted surface are deduced from Table VII. The differences between the skin horse-powers or resistances per 1 000 sq. ft. for ships of different lengths may be plotted separately as curves of correction.

Plates 3 to 6 of skin friction horse-power correction per 1 000 sq. ft. of wetted surface are examples of these derived curves, to be used for correcting the power when passing from one length of ship to another at the corresponding speeds (or speed of their 100-ft. model), or when reducing any ship to a 100-ft. model. Similar curves are used at experimental tank works for making the necessary correction when passing from the scale of a tank model to an actual ship.

It is only the skin frictional element of the horse-power that has to be corrected; the remainder varies as $l^{\cdot}$, and may be obtained directly by division. That is, as stated in the Introduction, the Law of Comparison applies to resistances other than frictional.

In analysing the results of progressive steam trials, or towing trials (*i.e.* trials measuring the tow-rope resistance at various speeds), the skin resistances are computed separately, and written in a column opposite the speeds. (See, for example, p. 205, trials of ferry steamer "Cincinnati.")

For each speed the total resistance – the skin frictional resistance = the residuary resistance.

When reducing the results of the progressive trial to the 100-ft. model, the skin resistances are corrected for friction, or calculated separately, while the residuary resistances are all reduced directly by dividing by l^3.

In the horse-power columns the only difference in the process is that the remainder (or H.P. left after deducting the skin H.P.) is divided by $l^{3\cdot5}$ instead of l^3.

For

$$l^{3\cdot5} = l^{3\frac{1}{2}} = l^{3+\frac{1}{2}} = l^3\sqrt{l}$$

and horse-power always = resistance × (0·003 070 7 × speed) (see Introduction).

Note.—The speeds on Plates 3, 4, 5, 6 are the speeds of 100-ft. models only. The skin correction, or difference of height between the ordinates of the various curves, is only applicable at the particular corresponding speed of the 100-ft. model at which it is taken. These plates give the amount of correction to be added to, or subtracted from, the power of the 100-ft. model when passing from a ship of any length to a 100-ft. model.

When extraordinary speeds are attained, the conversion to the 100-ft. model introduces values of the skin frictional H.P. per 1 000 ft. of W.S. outside of the curves we have drawn.

Given the progressive trial of a coasting steamer 218 × 32·8 × 9·72 ft. mean draught at trial, mentioned on p. 107.

Knots.	I.H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
7	232	182
8	332	190
9	493	182
10	720	172
10·1	765	166

Let us reduce this to a 100-ft. model. We have

$$l = 2\cdot18, \quad \sqrt{l} = 1\cdot476, \quad l^3 = 10\cdot36, \quad l^{3\cdot5} = 15\cdot29.$$

Dimensions :

$$100 \times 15\cdot06 \times 4\cdot46 \quad w = 0\cdot69.$$

$$D_m = \frac{1\,370}{l^3} = \frac{1\,370}{10\cdot36} = 132\cdot5 \text{ tons.}$$

Wetted surface (by Mumford's formula)

$$= (100 \times 15\cdot06 \times 0\cdot69) + (100 \times 4\cdot46 \times 1\cdot7)$$
$$= 1\,800 \text{ sq. ft.}$$

Corresponding speeds:

$$\frac{7}{1{\cdot}476},\quad \frac{8}{1{\cdot}476},\quad \frac{9}{1{\cdot}476},\quad \frac{10}{1{\cdot}476},\quad \frac{10{\cdot}1}{1{\cdot}476}$$

$$= 4{\cdot}75,\quad 5{\cdot}42,\quad 6{\cdot}1,\quad 6{\cdot}78,\quad 6{\cdot}85 \text{ knots.}$$

The skin H.P. and residuary H.P. are discussed on p. 36.

From Plate 3 we find that the difference of skin H.P. correction for passing from a 306-ft. ship to a 218-ft. ship is 0·25 per 1 000 sq. ft. of wetted surface.

The dimensions of the new ship are

$$306 \times 46 \times 13{\cdot}68 \text{ ft. mean draught, at trial.}$$

$$\text{Displacement} = 3\,800 \text{ tons.}$$

$$l^{3{\cdot}5} = l^3 \times \sqrt{l} = 28{\cdot}65 \times 1{\cdot}749 = 50{\cdot}1.$$

The larger ship is not so much affected by the weather.

At deeper draught we should expect a much better result. At the corresponding load draught (17 ft.), Admiralty constant about 210.

In the discussion on Naval-Constructor Taylor's paper, on the U.S. model basin, at the American Society of Naval Architects and Marine Engineers in 1900, Mr John Thom's formula was mentioned, and is certainly worthy of notice.

$$\text{I.H.P.} = \frac{D^{\frac{2}{3}} V^4}{\sqrt{E} \times \sqrt{d} \times c}$$

where D = displacement in tons.
V = speed in knots.
E = length of entrance in feet.
c = a constant (varying from 55 to 120).
E = L − (L × p).
p = prismatic coefficient.

For estimating speeds and powers of known vessels at their limiting economical speeds, this is a satisfactory formula to use, and the values of c do not vary much within ordinary limits.

CHAPTER V.

THE ADMIRALTY CONSTANT.

By the Law of Comparison we can derive the horse-power for a proposed steamer from the known performances of a "similar ship," if we have one. Proprietors of experimental tanks make similar ships (or models of them) whenever they require them, and try them in the tank. But if we have not a "similar ship" to work from, we may adopt one of two courses: (1) Still using the Law of Comparison, select a list of vessels as nearly similar to ours as we can obtain, plot their progressive speed and power curves on squared paper, and then decide where our vessel comes in. This method should be practised, if only because it leads to systematic handling of data. (2) We may try other methods, and formulæ, for determining the power, always keeping the principle of similitude in view. Among the formulæ in general use, the Admiralty constant comes first.

$$\text{I.H.P.} = \frac{D^{\frac{2}{3}}V^3}{C}$$

or

$$C = \frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$$

where D = displacement in tons.
V = speed in knots.
C = the "constant," or coefficient of performance.

The values of C, which will be found in tables and curves later, vary with the size of ship, being less for small ships than for large ones (Plate 39).

As a method for calculating power, the Admiralty formula, "adjusted as experience directs," is still the quickest and most universally used.

Experience shows that the decrease in the value of C for smaller vessels is due (in addition to the greater skin friction)

to the proportionately greater eddy-making resistance from rough surfaces, and to the greater effect of rough sea and wind on small ships. For a given ship the value depends upon the speed. The curve of $\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$ from a progressive trial almost always rises between low speeds and moderate speeds, and then falls away again between moderate speeds and high speeds. See Plates 4, 5, 23, showing typical curves of C.

Before beginning to calculate the power for a given ship, her salient features dominating resistance should be written down :—

I. Proportions :—

(1) The ratio of beam to length (B_m). The breadth of the 100-ft. model shows this immediately. The number of beams to length is the reciprocal, and is still preferred by some people.

(2) The ratio of draught to length. If the vessel is of light draught, then so much the worse for propulsion, especially if she is also very broad.

II. Fulness :—

(3) The block coefficient, mid-area coefficient, and prismatic coefficient.

III. Form :—

The longitudinal distribution of displacement, depending upon the shape of the curve of sectional areas and the water-line, especially of the fore body.

IV. Speed-length ratio $\frac{V}{\sqrt{L}}$:—

(4) The speed divided by the square root of the hundredth part of the length, or the speed divided by the square root of the length and multiplied by 10 = the corresponding speed of the 100-ft. model.

In ordinary merchant ships, fulness and form have a greater influence than proportions.

In fast passenger vessels and channel steamers, increase of fulness of displacement increases the resistance more than either of the above factors.

In torpedo craft and destroyer types, proportions become the principal factor.

Consider whether the speed proposed is higher or lower than the appropriate limit of speed for that vessel; if lower, she will be easy to drive, and a little more power will produce an appreciable extra speed; if higher, an increase of speed requires an undue increase of power. This appropriate limit of speed is

called the "Limiting Economical Speed." It is often taken as the speed at which the I.H.P. is varying as about the fourth power of the speed.

This point may be found by trial, by drawing tangents to the speed-power curve. (At higher speeds the I.H.P. may vary as the 7th or 10th or 11th or a still higher power of the speed.)

It may be found also by logarithms, as described on p. 88.

In our progressive trials the limiting economical speed is named and marked by an arrow, and on some of the curves of $\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$ we have shown its position by a dot in a circle.

Having settled these preliminaries for the proposed vessel and one or two other ships selected for comparison, examine all the available progressive curves of Admiralty constant, and after marking the position of our $\frac{\text{speed}}{\sqrt{l}}$ on one of these, read off the value of the constant, and apply the formula $\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.

After long practice the values given in the Tables of Steamship Data may be turned to some account, but only if considered strictly with regard to their ratios of speed to "limiting speed."

For estimating power for propulsion, and comparing and predicting performances, there are several other methods :—

(1) The Admiralty coefficient used with S.H.P., taking $\frac{\text{S.H.P.}}{\cdot 92} = \text{I.H.P.}$; thus,

$$\text{S.H.P.} = \frac{\Delta^{\frac{2}{3}}V^3 \times \cdot 92}{C}.$$

[For a reciprocating engine driving its own pumps, the ratio of S.H.P. to I.H.P. would be about ·855, and perhaps slightly less for small powers.]

(2) Admiralty "constant" system of notation :

$$\text{(c)} = \frac{\text{E.H.P.}}{\Delta^{\frac{2}{3}} \times V^3} \times 427\cdot 1.$$

(3) The Law of Comparison, where similar ships at similar speeds having equal propulsive coefficients, and l = the ratio of their linear dimensions, have their E.H.P.'s varying as certain functions of l,—the skin H.P. varying as $l^{3\cdot 415}$ and the residuary H.P. as $l^{3\cdot 5}$.

(4) Independent estimate, where the skin H.P. is calculated,

the residuary H.P. is obtained by the use of Taylor's contours, the air resistance is calculated, and percentages are added to provide for appendages, fulness of form, engine friction, and propeller waste.

(5) Model experiments, as described later in the book.

The Admiralty displacement constant $\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = C$ varies with shape and proportion of hull and with speed, and of course with weather and sea conditions. In the constant system of notation of results of experiments on models used at the British Admiralty experiment works, the values of the constant C depend only on shape and speed; size of vessel as a factor which would cause variation is eliminated. The value of C is expressed as a constant for "similar" forms at "corresponding speeds," whatever the absolute size of the vessel. The results are usually presented in the form of C curves for different K values, to a base of M, or to a base of ratio of length of entrance to length of run. These C curves may be regarded as curves of $\frac{\text{E.H.P.}}{V^3}$ for any ship of a fixed displacement.

The appearance of the formula $\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = C$ suggests that it is based upon certain assumptions.

These are enumerated in an article by Mr Peter Doig in *International Marine Engineering*, August 1911, who gave a diagram intended to apply to cases in which the ratio $\frac{\text{Length}}{\text{Beam}}$ is somewhere between 7·15 and 9·54, particularly fine vessels, mail steamers, channel steamers, high-speed yachts.

The assumptions are:—(1) That the resistance varies as the square (and consequently the power as the cube) of the speed; (2) that the ratio $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ is constant; and (3) that resistance at any particular speed is proportional to wetted surface, or two-thirds power of the displacement, to which wetted surface is itself approximately proportional.

TABLE XV.

Type.	Length in feet.	Screws.	Machinery.	Speed in knots.	Block co-efficient.
Coasters	200–300	Single	Reciprocating	10–15	·55–·68
Cargo vessels	200–300	Single	Reciprocating	8–12	·65–·85
	300–400	Single or twin	Reciprocating or geared turbine	9–14	·65–·85
	400–600	Single or twin	Reciprocating or geared turbine	10–17	·65–·85
Fine passenger	250–400	Single or twin	Reciprocating or geared turbine	15–22	·45–·60
	250–400	Twin or triple.	Direct turbine	20–25	·45–·55
Intermediate liners or mail steamers	400 ft. and upwards	Twin	Reciprocating	14–20	·60–·70
		Triple	Turbine or combination	16–20	·60–·65
Fast liners	500 ft. and upwards	Twin	Reciprocating or geared turbine	19–23	·55–·62
		Triple or quadruple	Direct turbine	20–26	·55–·62

Plate 6 applies to sea speeds on actual service, under more or less adverse weather conditions.

CHAPTER VI.

METHODS OF PRESENTING DIMENSIONS.

Example.—Mr R. E. Froude's 1904 Type 4, Series A, is a ship $325 \times 57 \times 22$ feet draught. Displacement = 6 048 tons. Block coefficient = ·521. (The block coefficient seems to figure out just under ·52.)

For comparing with other vessels, the dimensions may be expressed according to one or other of the following systems used in the literature of the subject:—

(1) By Mr R. E. Froude's Constant System of Notation used at the Admiralty Experiment Works, and used also at the National Physical Laboratory, and by Mr Luke, the length, breadth, draught, displacement, and block coefficient are all embodied in the three symbols—

$$\textcircled{M} = 5{\cdot}453, \quad \textcircled{B} = {\cdot}956, \quad \textcircled{D} = {\cdot}368.$$

The figures are the actual dimensions of ship multiplied by

$$\frac{{\cdot}305\,7}{(\text{Displacement in tons})^{\frac{1}{3}}}.$$

They may be regarded as the actual dimensions of an imaginary model of the ship, of one cubic foot displacement. (*Trans. Inst. Naval Architects*, 1888.) See also p. 73.

(2) As a 100-ft. model: thus, $100 \times 17{\cdot}54 \times 6{\cdot}78$. $\Delta = 176{\cdot}3$ tons.

Using our Table XIII on p. 41, $l = 3{\cdot}25$, $l^3 = 34{\cdot}33$. $\frac{6\,048}{34{\cdot}33} = 176{\cdot}3$. The breadth and draught are percentages of the length, and the displacement 176·3 is the same as Mr Taylor's $\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.

(3) By bringing it to a standard displacement of 10 000 tons.

Here we have $\frac{10\,000}{176{\cdot}3} = l^3$, $\therefore l = 3{\cdot}842\,5$. Length = 384·25.

$\therefore$ the dimensions are $384{\cdot}25 \times 67{\cdot}4 \times 26{\cdot}08$, with block coefficient = ·52, displacement = 10 000 tons.

(4) By bringing it to a standard length of 400 ft., a method employed by Mr Baker, and by Mr R. E. Froude in earlier papers. $400 \times 70{\cdot}12 \times 27{\cdot}12$. Displacement = 11 300 tons.

(5) Mr Taylor's notation, which we have adopted to some extent in our tables, pp. 102, 358. L = 325. $\frac{\text{B}}{\text{H}} = 2{\cdot}59$. Beam as percentage of length = 17·54. $\frac{\Delta}{\left(\frac{100}{\text{L}}\right)^3} = 176{\cdot}3$.

The following shows the application of Mr Froude's Constant System of Notation to Mr Taylor's data :—

Let V = speed in knots.
r = resistance in lbs. in fresh water.
δ = displacement in lbs. in fresh water.
L = length in feet between perpendiculars.
S = wetted surfaces in square feet.

(Mr Taylor's models were run naked, *i.e.* without appendages such as bossings, etc.)

In the case of Model No. 1107 :

$$K = \frac{v}{\delta^{\frac{1}{6}}} \times 2{\cdot}074 \qquad K = \frac{v}{3{\cdot}619} \times 2{\cdot}074$$

$$C = \frac{r}{\delta^{\frac{2}{3}} v^2} \times 232{\cdot}5 \qquad C = \frac{r}{171{\cdot}7 v^2} \times 232{\cdot}5 = 1{\cdot}354 \frac{r}{v^2}$$

$$L = \frac{\text{V}}{\sqrt{\text{L}}} \times 1{\cdot}055\,2 \qquad L = \frac{\text{V}}{4{\cdot}472} \times 1{\cdot}055\,2 = \text{V} \times {\cdot}236\,1$$

$$L \text{ also} = \frac{\text{K}}{\sqrt{\text{M}}} \qquad M = \frac{20}{13{\cdot}104} \times 3{\cdot}966$$

$$M = \frac{\text{L}}{\delta^{\frac{1}{3}}} \times 3{\cdot}966 \qquad {}^*S = \frac{70{\cdot}7}{171{\cdot}7} \times 15{\cdot}73 \text{ to } \frac{72{\cdot}4}{171{\cdot}7} \times 15{\cdot}73$$

$$S = \frac{\text{S}}{\delta^{\frac{2}{3}}} \times 15{\cdot}73 \qquad B = \frac{2{\cdot}795}{13{\cdot}104} \times 3{\cdot}966$$

$$B = \frac{\text{Beam}}{\delta^{\frac{1}{3}}} \times 3{\cdot}966 \qquad D = \frac{1{\cdot}118}{13{\cdot}104} \times 3{\cdot}966$$

$$D = \frac{\text{Draught}}{\delta^{\frac{1}{3}}} \times 3{\cdot}966$$

(Note that the "constants" are in italics.)

* Wetted surfaces of Taylor's models a. 1107 and d. 1092, 1·8 per cent. and 4 per cent. in excess of Mumford's wetted surfaces respectively, Mumford's wetted surface being 69·45 sq. ft. Mr Taylor's values of C in his formula and curves for wetted surface are for naked models.

The following are Mr R. E. Froude's constants:—

Let V = speed in knots; v = do. in hundreds of ft. per min.
R = resistance in tons in salt water; r = do. in lbs. in fresh water.
Δ = displacement in tons in salt water; δ = do. in lbs. in fresh water.
L = length in feet between perpendiculars.
S = wetted skin area in square feet.

Then—

(1) The "Speed Constant" (*K*), which expresses speed relatively to displacement to the one-sixth power,

$$= \frac{V}{\Delta^{\frac{1}{6}}} \times \cdot 583\,4 = \frac{v}{\delta^{\frac{1}{6}}} \times 2{\cdot}074.$$

(2) The "Resistance Constant" (*C*), which expresses resistance relatively to the square of the speed multiplied by the two-thirds power of the displacement,

$$= \frac{R}{\Delta^{\frac{2}{3}} V^2} \times 2\,938 = \frac{r}{\delta^{\frac{2}{3}} v^2} \times 232{\cdot}5$$

$$= \frac{\text{E.H.P.}}{\Delta^{\frac{2}{3}} V^3} \times 427{\cdot}1.$$

(3) The "Length-Speed-Constant" (*L*), which expresses speed relatively to the square root of the length,

$$= \frac{V}{\sqrt{L}} \times 1{\cdot}055\,2 = \frac{v}{\sqrt{L}} \times 010\,41.$$

The following indicates the method of obtaining the numerical value of the "Length-Speed Constant" (capital L, italics, in a circle):—

$$(L) = \frac{\text{Velocity of ship}}{\text{Velocity of wave of length} = \text{half length of ship}}$$

$$= \frac{\text{V in ft. per sec.}}{\sqrt{\dfrac{g \cdot \frac{L}{2}}{2\pi}}} = \frac{v \text{ in hundreds of ft. per min.} \times \frac{100}{60}}{\sqrt{\dfrac{gL}{4\pi}}}$$

$$= \frac{v}{\sqrt{L}} \times \frac{100}{60} \times \sqrt{\frac{4\pi}{32{\cdot}2}} = \frac{v}{\sqrt{L}} \times \frac{100}{60} \sqrt{\cdot 390\,417}$$

$$= \frac{v}{\sqrt{L}} \times \frac{\cdot 624\,8}{\cdot 6} = 1{\cdot}041 \frac{v}{\sqrt{L}}.$$

Or, from another point of view,

$$\textcircled{L} = 1{\cdot}055\,2\frac{V}{\sqrt{L}}, \text{ where V is speed in knots.}$$

1 knot = 101⅓ ft. per min. = 1·013 3 hundreds of ft. per min.
If V is in hundreds of ft. per min.

$$\text{V in knots} = \frac{v \text{ in hundreds of ft. per min.}}{1{\cdot}013\,3}$$

$$\therefore \textcircled{L} = \frac{v \text{ in hundreds of ft. per min.}}{\sqrt{L}} \times \frac{1{\cdot}055\,2}{1\,013\,3}$$

$$= \frac{1{\cdot}041v}{\sqrt{L}}.$$

(4) The "Length Constant" $\textcircled{M}$, the ratio of the length of ship to the side of the cube containing the displacement,

$$= \frac{L}{\Delta^{\frac{1}{3}}} \times {\cdot}305\,7 = \frac{L}{\delta^{\frac{1}{3}}} \times 3{\cdot}966.$$

(5) Equally, the constant for any linear dimension (*e.g.* $\textcircled{B}$ or $\textcircled{D}$ for beam or draught), the ratio of the beam or draught of ship to the side of the cube containing the displacement,

$$= \frac{\text{Dimension}}{\Delta^{\frac{1}{3}}} \times {\cdot}305\,7.$$

(6) The "Skin Constant" $\textcircled{S}$ expresses wetted surface relatively to the two-thirds power of the displacement,

$$= \frac{S}{\Delta^{\frac{2}{3}}} \times {\cdot}093\,46 = \frac{S}{\delta^{\frac{2}{3}}} \times 15{\cdot}73.$$

Note also that $\textcircled{L} = \dfrac{K}{\sqrt{M}}$.

In Mr R. E. Froude's "Constant" system the constants are the same for the model as for the ship.

Taking dimensions from the following example, Sadler, *Trans. American Society Naval Arch. and Marine Engineers*, 1915, we can show that Mr R. E. Froude's "Skin Constant" $\textcircled{S}$ has the same value for ship and for model.

(1) Type 2 (*b*) as a 400-ft. ship. $\Delta = 6\,150$.

Wetted surface by Mumford's formula :—

$$
\begin{aligned}
400 \times 50 \times \cdot 537 &= 10\,750 \\
400 \times 20 \times 1\cdot 7 &= 13\,600 \\
\hline
S &= 24\,350
\end{aligned}
$$

$$
\begin{aligned}
s &= \frac{S}{\Delta^{\frac{2}{3}}} \times \cdot 093\,46 \\
&= \frac{24\,350}{335\cdot 67} \times \cdot 093\,46 \\
&= 6\cdot 79.
\end{aligned}
$$

(2) Type 2 (*b*) as a 100-ft. ship. $\Delta = 96$.
Wetted surface by Mumford's formula :—

$$
\begin{aligned}
100 \times 12\cdot 5 \times \cdot 537 &= 671 \\
100 \times 5\cdot 0 \times 1\cdot 7 &= 850 \\
\hline
S &= 1\,521
\end{aligned}
$$

$$
\begin{aligned}
s &= \frac{1\,521}{20\cdot 95} \times \cdot 093\,46 \\
&= 6\cdot 79.
\end{aligned}
$$

The method of applying these constants is very simple. All it entails is multiplying the ordinates, or c values, by the constants given on p. 71, thus giving us the E.H.P. for any length of ship, the skin friction correction being part of the c value. The constant system lends itself better than any other to research work, and can be applied by practical ship designers. It has the merit of presenting the Admiralty coefficient, favoured by engineers, disguised somewhat, and inverted, but still the language in which they are accustomed to think, and varying characteristically, as they know it does vary. Mr R. E. Froude's paper "On the Constant System of Notation of Results of Experiments on Models used at the Admiralty Experiment Works," read before the Institution of Naval Architects in 1888, describes the method of expressing the values of the resistance constant c, and the speed constant K, constant for "similar" forms at "corresponding speeds," whatever the absolute size. The resistance constant is virtually the formula $\frac{\Delta^{\frac{2}{3}} V^3}{\text{Horse-power}}$ turned upside down for the sake of having the horse-power in the numerator,

as in this way the skin friction correction and other constituents of the resistance can be apportioned for the case under consideration.

Mr R. E. Froude's 1904 paper to the Inst. N.A. gave an account of experiments with six different sets of lines, varied in proportion by independent variation of length, beam, and draught scales. Each set of lines, or parent form, or "type," was subjected to variations in proportion, consisting chiefly of variations in length scale relatively to cross-section scale, the proportion of beam to draught remaining unaltered. This variation in length proportion was represented in the models as a variation in cross-section scale, length remaining unaltered, giving a range of variation in proportion extending from 2 500 tons up to 10 500 tons for the 350 ft. length of Type 1, A, with $\frac{\text{Beam}}{\text{Draught}} = \frac{57}{22}$, the original 6 100 tons forming one of the intermediate gradations. Stating this range of variation in the "constant" system of notation, the range is from an M value of 7·884 corresponding to 2 500 tons to 4·886 for 10 500 tons. M = the ratio of the length of ship to the side of the cube containing the displacement.

Another grade was tried, B, with $\frac{\text{Beam}}{\text{Draught}} = \frac{66}{19}$, for 350 ft. length and 6 100 tons displacement, in which six "types" of form were tried, the range of length proportion being from 1 250 up to 7 750 tons, corresponding to an M value range of from 9·933 to 5·407. Resistance was expressed in C values, *i.e.* the relation of $(\text{speed})^2 \times (\text{displacement})^{\frac{2}{3}}$ and for constant engine and propeller efficiency. The speed constant used was K, which expresses speed relatively to $(\text{displacement})^{\frac{1}{6}}$. For ships of the same model, at "corresponding" speeds, C and K are independent of absolute size (apart from skin friction correction). For each value of K there was a curve of C plotted to a base of M, and these were termed "Iso-K" curves. For every K value there were twelve "Iso-K" curves (one for each of the six types, each of the two series A and B). Twenty-nine different values of K were taken, each appropriating a separate diagram. Skin friction correction curves were plotted under the C ordinates of the "Iso-K" curves. On each "Iso-K" diagram there was a curve for converting C into E.H.P., and another for converting K into speed; and one for converting the constants into actual ship dimensions.

Mr R. E. Froude's 1904, Type 4, Series A. K = 2·8. $\frac{\text{Beam}}{\text{Draught}} = \frac{57}{22} = 2{\cdot}59$. Speed = 20·5 knots. Δ = 6 048 tons. Derived by the "constant" system from the type ship in the third line.

(M)	Dimensions in feet.			Coefficients.			Immersed midship area.
	Length.	Beam.	Draught.	Block.	Mid area.	Pris-matic.	
4·6	274	61·9	23·85	·524	·877 5	·598	1 293
5·0	298	59·6	23	·518	·877 5	·590	1 200
(type ship) 5·453	325	57	22	·521	·877 5	·594	1 100
6·0	358	54·4	21	·517	·877 5	·589	1 001
6·6	393·5	51·6	19·6	·530 5	·877 5	·605	887
7·0	418	50·3	19·33	·52	·877 5	·593	854
7·4	441	48·6	18·78	·525	·877 5	·600	801

If the reader applies for himself the formula $\frac{\Delta^{\frac{2}{3}} V^3}{\text{Horse-power}}$ for a ship and for its model, he will be met with the difficulty of making the values agree, but with Mr Froude's method the (c) values determined from experiments on a model can be very conveniently corrected for a ship by deducting from the (c) value for the model the net value $F_M - F_S$, where F_M = the skin friction term in the (c) value for the model, and F_S = the skin friction term in the (c) value for the ship.

$$F_M - F_S = (O_M - O_S)\,SL^{-\cdot 175} \qquad O \propto L^{-\cdot 175}.$$

Mr R. E. Froude's 1904, Type 4, Series A, modified for comparison with Taylor's Standard Series, (1) by increasing the length from b.p. to l.w.l. to suit Taylor's cruiser stern, and (2) by altering the beam ÷ draught ratio to correspond with Taylor's midship section ratio ·926. Froude's ship lengths are lengths b.p., the form having the advantages which accompany the cruiser stern. Taylor's length must therefore be increased by an amount judged from scaling the profile.

$$\left.\begin{aligned}\frac{\text{Beam}}{\text{Draught}} \times \frac{\cdot 926}{\cdot 877\ 5} = \text{new ratio } \frac{\text{Beam}}{\text{Draught}} &= \frac{57}{22} \times \frac{\cdot 926}{\cdot 877\ 5} = 2\cdot 735. \\ &= 2\cdot 59 \times \frac{\cdot 926}{\cdot 877\ 5} = 2\cdot 735\end{aligned}\right\} \Delta = 6\ 048 \text{ tons.} \quad \text{Speed} = 20\cdot 5 \text{ knots.}$$

Neither Froude's 1904 Series nor Taylor's Standard Series have parallel body.

Ⓜ	Ⓒ scaled from the diagrams corrected for skin friction.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Immersed midship area.	Modified dimensions. Length.*	Modified dimensions. Beam.	Modified dimensions. Draught.*	$\frac{\text{Length}}{\text{Beam}}$.	Beam as percentage of length.	Coefficients. Block.	Coefficients. Prismatic.	Coefficients. Mid area.	Approximate wetted skin.	Taylor's residuary resistance lbs. per ton Δ.	$\frac{V}{\sqrt{L}}$.	Resid. resistance in lbs. per ton Δ from Froude's Ⓒ.	E.H.P.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.
4·6	1·753	258·3	1 293	286	61·9	22·64	4·625	21·62	·528	·57	·926	20 360	..	1·212	..	11 740	121·7
5·0	1·208	201	1 200	311	59·6	21·82	5·21	19·19	·522	·564	·926	21 240	15·795	1·164	13·71	8 100	176·5
5·453	·962	155	1 100	339	57	20·86	5·95	16·81	·525 5	·568	·926	22 150	9·46	1·115	9·04	6 450	222
6·0	·841 1	116	1 001	373·5	54·4	19·9	6·87	14·56	·524	·566	·926	23 270	6·438	1·063	6·55	5 640	253·5
6·6	·779	87·7	887	410	51·6	18·9	7·95	12·6	·529	·571	·926	24 370	4·935	1·014	5·08	5 220	274
7·0	·753	73·4	854	435	50·3	18·4	8·65	11·56	·526	·569	·926	25 110	3·863	·984	4·38	5 045	283·2
7·4	·737 6	62·1	801	460	48·6	17·8	9·46	10·57	·531	·574	·926	25 810	3·194	·956	3·94	4 940	290

For calculating $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$, the value of $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ has been taken as $= \cdot 50$.

* (1) The lengths are Froude's lengths ÷ ·96 to bring b.p. to l.w.l. (2) The draughts are those obtained by altering the beam ÷ draught ratio of Froude's ships to compare with Taylor's ships, which have a midship section coefficient of ·926.

The values of O for various lengths of ship are given in the table below.

TABLE XVI.—TABLE OF VALUES OF O FOR VARIOUS LENGTHS.

Length in feet.	Value of "O."	Length in feet.	Value of "O."	Length in feet.	Value of "O."	Length in feet.	Value of "O."
8	·140 90	80	·089 87	350	·075 25	620	·070 25
9	·137 34	90	·088 40	360	·075 0	640	·070 0
10	·134 09	100	·087 16	380	·074 57	660	·069 75
12	·128 58	120	·085 11	400	·074 12	680	·069 52
14	·124 06	140	·083 51	420	·073 71	700	·069 31
16	·120 35	160	·082 19	440	·073 31	720	·069 08
18	·117 27	180	·081 08	450	·073 12	740	·068 85
20	·114 70	200	·080 12	460	·072 94	760	·068 61
25	·109 76	220	·079 25	480	·072 57	780	·068 40
30	·105 90	240	·078 5	500	·072 19	800	·068 19
35	·102 82	250	·078 14	520	·071 83	820	·068 0
40	·100 43	260	·077 8	540	·071 49	840	·067 8
45	·098 39	280	·077 15	550	·071 32	860	·067 6
50	·096 64	300	·076 55	560	·071 15	880	·067 4
60	·093 80	320	·076 04	580	·070 83	900	·067 22
70	·091 64	340	·075 5	600	·070 51		

TABLE XVII.—MULTIPLIERS FOR MR R. E. FROUDE'S SKIN FRICTION COEFFICIENTS.

Functions of the Length-Speed-Constant L.

L	$L^{-\cdot175}$	L	$L^{-\cdot175}$	L	$L^{-\cdot175}$	L	$L^{-\cdot175}$
·10	1·496 2	·52	1·121	·90	1·018 6	1·28	·957 5
·15	1·396	·53	1·118	·91	1·017	1·29	·956 7
·16	1·379	·54	1·114	·92	1·015	1·30	·955 12
·17	1·365	·55	1·111	·93	1·013	1·31	·954
·18	1·352	·56	1·108	·94	1·011 6	1·32	·952 5
·19	1·338	·57	1·104	·95	1·009 5	1·33	·951 5
·20	1·325	·58	1·101	·96	1·008	1·34	·950
·21	1·313	·59	1·098	·97	1·006 5	1·35	·948 4
·22	1·303	·60	1·093 5	·98	1·004 2	1·36	·947 2
·23	1·293	·61	1·091	·99	1·002 5	1·37	·946 3
·24	1·284	·62	1·088	1·0	1·000 0	1·38	·945
·25	1·274	·63	1·084	1·01	·998 2	1·39	·943 3
·26	1·266	·64	1·081	1·02	·996 8	1·40	·942 82
·27	1·257	·65	1·079	1·03	·995	1·41	·941
·28	1·25	·66	1·076	1·04	·993	1·42	·939 2
·29	1·243	·67	1·073	1·05	·991 6	1·43	·939
·30	1·234 5	·68	1·07	1·06	·990	1·44	·938
·31	1·227	·69	1·067	1·07	·988	1·45	·937
·32	1·221	·70	1·064 4	1·08	·986 6	1·46	·936
·33	1·214	·71	1·061	1·09	·985	1·47	·935
·34	1·208	·72	1·059	1·10	·983 46	1·48	·933 5
·35	1·203	·73	1·057	1·11	·981 7	1·49	·932
·36	1·196	·74	1·054	1·12	·980	1·50	·931 50
·37	1·19	·75	1·051	1·13	·978 2	1·51	·931
·38	1·185	·76	1·048	1·14	·977	1·52	·929
·39	1·18	·77	1·046	1·15	·976	1·53	·928
·40	1·173 9	·78	1·043	1·16	·974	1·54	·927
·41	1·169	·79	1·041	1·17	·972 5	1·55	·926
·42	1·164	·80	1·039 8	1·18	·971 3	1·56	·925
·43	1·159	·81	1·037	1·19	·969 6	1·57	·924
·44	1·154	·82	1·035	1·20	·968 60	1·58	·923
·45	1·15	·83	1·032 5	1·21	·966 7	1·59	·922
·46	1·145	·84	1·030 5	1·22	·965 6	1·60	·921 04
·47	1·141	·85	1·028 5	1·23	·964	1·61	·920
·48	1·137	·86	1·026 5	1·24	·962 6	1·62	·919
·49	1·133	·87	1·024 5	1·25	·961 8	1·63	·918
·50	1·129	·88	1·023	1·26	·960 6	1·64	·917
·51	1·125	·89	1·020 5	1·27	·959	1·65	·916

TABLE XVII.—MULTIPLIERS FOR MR R. E. FROUDE'S SKIN FRICTION COEFFICIENTS—*continued.*

Functions of the Length-Speed-Constant (L).

(L)	$L^{-.175}$	(L)	$L^{-.175}$	(L)	$L^{-.175}$	(L)	$L^{-.175}$
1·66	·915	2·04	·883 3	2·42	·856 8	3·5	·803 13
1·67	·914	2·05	·882 4	2·43	·856	3·6	·799 18
1·68	·913	2·06	·881 7	2·44	·854	3·7	·795 36
1·69	·912	2·07	·881 0	2·45	·854 9	3·8	·791 66
1·70	·911 32	2·08	·880	2·46	·854 2	3·9	·788 07
1·71	·910	2·09	·879	2·47	·853 6	4·0	·784 58
1·72	·909 2	2·10	·878 24	2·48	·853	4·1	·781 20
1·73	·908 5	2·11	·878	2·49	·852 4	4·2	·777 91
1·74	·907 5	2·12	·877	2·50	·851 84	4·3	·774 72
1·75	·906 6	2·13	·876 5	2·51	·851 2	4·4	·771 61
1·76	·905 6	2·14	·876	2·52	·850 6	4·5	·768 58
1·77	·905	2·15	·875	2·53	·850	4·6	·765 63
1·78	·903 6	2·16	·874 2	2·54	·849 5	4·7	·762 75
1·79	·902 6	2·17	·873 6	2·55	·848 9	4·8	·759 95
1·80	·902 25	2·18	·872 7	2·56	·848 3	4·9	·757 21
1·81	·901	2·19	·871 9	2·57	·847 9	5·0	·754 54
1·82	·900	2·20	·871 12	2·58	·847 2	5·1	·751 93
1·83	·899	2·21	·870 6	2·59	·846 6	5·2	·749 38
1·84	·898 5	2·22	·870	2·60	·846 02	5·3	·746 88
1·85	·897 5	2·23	·869	2·61	·845 5	5·4	·744 44
1·86	·896 9	2·24	·868 6	2·62	·845	5·5	·742 06
1·87	·896	2·25	·868	2·63	·844 4	5·6	·739 72
1·88	·895	2·26	·867	2·64	·843 9	5·7	·737 43
1·89	·894 4	2·27	·866 5	2·65	·843 2	5·8	·735 19
1·90	·893 75	2·28	·866	2·66	·842 7	5·9	·732 995
1·91	·892 7	2·29	·865	2·67	·842 1	6·0	·730 84
1·92	·892 3	2·30	·864 37	2·68	·841 6	6·1	·728 73
1·93	·891	2·31	·863 7	2·69	·841	6·2	·726 66
1·94	·890 5	2·32	·863	2·70	·840 45	6·3	·724 63
1·95	·890	2·33	·862 5	2·75	·837 9	6·4	·722 63
1·96	·889	2·34	·861 9	2·80	·835 12	6·5	·720 68
1·97	·888 5	2·35	·861 2	2·85	·832 5	6·6	·718 75
1·98	·887 6	2·36	·860 5	2·90	·830 00	6·7	·716 86
1·99	·886 6	2·37	·860	3·0	·825 09	6·8	·715 01
2·00	·885 77	2·38	·859 3	3·1	·820 37	6·9	·713 18
2·01	·885	2·39	·858 6	3·2	·815 83	7·0	·711 39
2·02	·884 6	2·40	·857 95	3·3	·811 45	7·1	·709 63
2·03	·884	2·41	·857 3	3·4	·807 22	7·1	·707 89

TABLE XVIII.—POWERS OF THE SPEED FOR SHIPS IN SALT WATER.

V	$V^{1.83}$	$V^{2.83}$	$V^{1.825}$	$V^{2.825}$	V	$V^{1.83}$	$V^{2.83}$	$V^{1.825}$	$V^{2.825}$
1	1	1	1	1	4·5	15·56	70	15·5	69·7
1·1	1·19	1·31	1·19	1·31	4·6	16·3	75	16·2	74·5
1·2	1·40	1·68	1·396	1·673	4·7	17	80	16·9	79·4
1·3	1·62	2·1	1·614	2·1	4·75	17·37	82·5	17·2	81·7
1·4	1·853	2·696	1·846	2·583	4·8	17·7	85	17·5	84
1·5	2·1	3·15	2·05	3·08	4·9	18·4	90	18·2	89·1
1·6	2·36	3·79	2·36	3·78	5·0	19	95	18·86	94·3
1·7	2·64	4·48	2·63	4·47	5·1	19·8	100	19·54	99·6
1·75	2·78	4·89	2·78	4·86	5·2	20·5	106	20·32	105·7
1·8	2·93	5·28	2·92	5·26	5·25	20·8	109	20·7	108·8
1·9	3·23	6·14	3·22	6·12	5·3	21·3	112	20·94	111·0
2·0	3·56	7·11	3·54	7·09	5·4	22	118	21·7	117·1
2·1	3·89	8·16	3·88	8·15	5·5	22·6	124	22·44	123·4
2·2	4·23	9·3	4·22	9·29	5·6	23·5	131	23·1	129·2
2·25	4·42	9·95	4·4	9·9	5·7	24·2	138	23·9	136·1
2·3	4·58	10·52	4·57	10·51	5·75	24·6	141·5	24·3	139·6
2·4	4·97	11·93	4·95	11·88	5·8	25	145	24·7	143·1
2·5	5·35	13·38	5·15	12·88	5·9	25·8	152	25·6	151
2·6	5·76	15	5·56	14·47	6·0	26·5	159	26·31	157·8
2·7	6·17	16·68	6·0	16·2	6·1	27·5	167	27·1	165·1
2·75	6·36	17·5	6·33	17·4	6·2	28·3	175	28·0	173·5
2·8	6·61	18·5	6·54	18·3	6·25	28·6	179	28·3	176·9
2·9	7·01	20·33	6·99	20·26	6·3	29·2	183	28·7	180·8
3·0	7·48	22·42	7·42	22·2	6·4	30·0	191	29·6	189·4
3·1	7·95	24·65	7·91	24·43	6·5	30·8	200	30·34	197
3·2	8·41	26·9	8·34	26·68	6·6	31·7	209	31·25	206
3·25	8·63	28·04	8·59	27·92	6·7	32·6	218	32·24	216
3·3	8·89	29·3	8·82	29·1	6·75	33	222·5	32·5	219·5
3·4	9·42	32	9·34	31·75	6·8	33·6	227	33·1	225
3·5	9·91	34·7	9·86	34·52	6·9	34·4	236	33·9	234
3·6	10·44	37·6	10·33	37·2	7·0	35·2	246	34·85	244
3·7	11	41	10·9	40·4	7·1	36·3	256	35·8	254
3·75	11·33	42·5	11·09	41·55	7·2	37·2	267	36·7	264·2
3·8	11·52	44	11·43	43·5	7·25	37·6	272·5	37·2	269·7
3·9	12·1	47	12·0	46·8	7·3	38·2	278	37·7	275·2
4·0	12·75	51	12·31	50·2	7·4	39·2	289	38·6	285·7
4·1	13·25	54	13·15	53·9	7·5	40	300	39·7	297·9
4·2	13·85	58	13·7	57·5	7·6	41·1	311	40·6	308·5
4·25	14·11	60	14·0	59·5	7·7	42·1	323	41·6	320·5
4·3	14·41	62	14·3	61·45	7·75	42·5	329	41·9	324·5
4·4	15·03	66	14·9	65·55	7·8	43	335	42·5	331·6

TABLE XVIII.—POWERS OF THE SPEED FOR SHIPS IN SALT WATER—*continued.*

V	$V^{1\cdot83}$	$V^{2\cdot83}$	$V^{1\cdot825}$	$V^{2\cdot825}$	V	$V^{1\cdot83}$	$V^{2\cdot83}$	$V^{1\cdot825}$	$V^{2\cdot825}$
7·9	44	347	43·5	343·8	11·3	84·7	956	83·4	943
8·0	45	360	44·47	355·8	11·4	86	980	84·9	968
8·1	46·2	373	45·6	370	11·5	87·4	1 004	86·2	992
8·2	47·2	386	46·5	381·5	11·6	88·9	1 029	87·7	1 019
8·25	47·6	392·5	46·9	387	11·7	90·1	1 054	89	1 041
8·3	48·1	399	47·6	394·4	11·75	90·9	1 067	89·7	1 055
8·4	49·2	413	48·8	409·5	11·8	91·8	1 080	90·4	1 067
8·5	50·3	427	49·7	423	11·9	93	1 106	91·9	1 093
8·6	51·4	441	50·7	426	12·0	94·4	1 133	93·21	1 118
8·7	52·5	456	51·6	449	12·1	96	1 160	94·9	1 149
8·75	52·9	463·5	52·4	458·3	12·2	97·3	1 187	96	1 171
8·8	53·6	471	52·8	464·5	12·25	98·2	1 201	96·5	1 181
8·9	54·6	486	54	480·5	12·3	98·9	1 215	96·9	1 191
9·0	55·7	502	55·14	496·2	12·4	100·2	1 243	99	1 229
9·1	56·9	518	56·2	511·5	12·5	101·8	1 271	100·3	1 254
9·2	58	534	57·4	528	12·6	103·2	1 300	101·8	1 281
9·25	58·6	542·5	58	536	12·7	104·9	1 330	103·1	1 310
9·3	59·4	551	58·6	545	12·75	105·6	1 345	104	1 327
9·4	60·5	568	59·8	561·6	12·8	106·2	1 360	104·7	1 340
9·5	61·6	585	61	579·5	12·9	107·8	1 390	106·1	1 370
9·6	62·8	602	62	595	13·0	109·4	1 421	107·8	1 402
9·7	64	620	63·1	613	13·1	110·9	1 452	109·3	1 432
9·75	64·6	630	63·7	621	13·2	112·4	1 483	110·9	1 462
9·8	65·1	639	64·2	629	13·25	113·1	1 499	111·5	1 477
9·9	66·5	657	65·4	646	13·3	114	1 515	112·4	1 495
10·0	67·6	676	66·83	668·3	13·4	115·6	1 548	114	1 529
10·1	69	695	68	686	13·5	117	1 581	115·5	1 560
10·2	70·2	715	69·2	706	13·6	118·8	1 614	117	1 591
10·25	70·7	725	70	719	13·7	120·3	1 648	118·9	1 629
10·3	71·6	735	70·6	728	13·75	121·1	1 665	119·4	1 642
10·4	72·8	755	71·9	747	13·8	122	1 682	120·2	1 660
10·5	73·9	776	73·1	769	13·9	123·6	1 717	121·9	1 692
10·6	75·3	797	74·4	789	14·0	125·2	1 752	123·5	1 729
10·7	76·8	819	75·6	810	14·1	126·8	1 788	125	1 761
10·75	77·3	830	76·2	820	14·2	128·4	1 824	127·4	1 810
10·8	78	841	77	831	14·25	129·3	1 842	128	1 824
10·9	79·4	863	78·3	853	14·3	130	1 860	128·6	1 839
11·0	80·5	885	79·53	874·8	14·4	131·7	1 897	130·2	1 876
11·1	82	908	80·9	897	14·5	133·5	1 935	131·9	1 911
11·2	83·2	932	82·1	920	14·6	135·1	1 973	133·5	1 949
11·25	83·9	944	82·9	933	14·7	136·9	2 012	135	1 985

TABLE XVIII.—POWERS OF THE SPEED FOR SHIPS IN SALT WATER—*continued.*

V	$V^{1.83}$	$V^{2.83}$	$V^{1.825}$	$V^{2.825}$	V	$V^{1.83}$	$V^{2.83}$	$V^{1.825}$	$V^{2.825}$
14·75	137·9	2 031	136	2 006	18·2	202·4	3 681	199	3 620
14·8	138·6	2 051	136·8	2 022	18·25	203·5	3 710	200	3 650
14·9	140·2	2 090	138·4	2 061	18·3	204·1	3 739	201	3 680
15·0	142	2 130	140	2 101	18·4	206·2	3 797	203	3 740
15·1	143·7	2 170	142	2 142	18·5	208·4	3 856	205·1	3 800
15·2	145·5	2 211	143·8	2 184	18·6	210·4	3 915	207·2	3 860
15·25	146·4	2 231	144·5	2 205	18·7	212·8	3 975	209·3	3 912
15·3	147·1	2 252	145·4	2 223	18·75	214	4 010	210·3	3 942
15·4	149	2 294	147·1	2 266	18·8	214·8	4 035	211·5	3 975
15·5	150·8	2 337	149	2 310	18·9	216·8	4 096	213·5	4 037
15·6	152·5	2 380	150·5	2 348	19·0	219	4 158	215·6	4 097
15·7	154·4	2 423	152·3	2 392	19·1	221·2	4 220	217·7	4 155
15·75	155·3	2 445	153·1	2 415	19·2	223·2	4 283	219·7	4 213
15·8	156·1	2 467	154	2 433	19·25	224·5	4 314	220·7	4 250
15·9	158	2 512	155·9	2 477	19·3	225·2	4 346	221·8	4 280
16·0	159·9	2 557	157·5	2 521	19·4	227·6	4 410	223·8	4 340
16·1	161·7	2 602	159·4	2 568	19·5	229·6	4 475	225·9	4 400
16·2	163·4	2 648	161·3	2 615	19·6	231·8	4 540	228	4 466
16·25	164·4	2 672	162·1	2 638	19·7	234	4 606	230	4 535
16·3	165·1	2 695	163·1	2 660	19·75	235	4 640	231	4 560
16·4	167·2	2 742	164·9	2 702	19·8	236	4 673	232·1	4 600
16·5	169	2 789	166·7	2 750	19·9	238·2	4 740	234·3	4 660
16·6	170·7	2 837	168·4	2 796	20·0	240·5	4 808	236·8	4 735
16·7	173	2 886	170·2	2 942	20·1	242·9	4 876	238·7	4 800
16·75	173·2	2 910	171·2	2 870	20·2	244·8	4 945	241	4 860
16·8	174·8	2 935	172·1	2 893	20·25	246	4 980	242·1	4 908
16·9	176·9	2 985	174	2 940	20·3	247·6	5 015	243·1	4 940
17·0	178·5	3 035	176	2 992	20·4	249·3	5 085	245·3	5 002
17·1	180·7	3 086	177·8	3 040	20·5	251·6	5 156	247·7	5 085
17·2	182·4	3 137	179·4	3 085	20·6	253·9	5 227	250	5 150
17·25	183·5	3 163	180·3	3 111	20·7	255·9	5 299	252	5 220
17·3	184·4	3 189	181·3	3 140	20·75	257	5 335	253·2	5 255
17·4	186·6	3 242	183·2	3 190	20·8	258·4	5 372	254·3	5 295
17·5	188·3	3 295	185·1	3 240	20·9	260·6	5 445	256·6	5 360
17·6	190·3	3 348	187·1	3 292	21·0	262·9	5 519	258·8	5 435
17·7	192·5	3 402	189	3 346	21·1	265·2	5 594	261·2	5 512
17·75	193·3	3 430	190	3 371	21·2	267·8	5 669	263·5	5 590
17·8	194·2	3 457	191	3 400	21·25	268·8	5 707	264·6	5 630
17·9	196·2	3 512	193	3 445	21·3	269·9	5 745	265·7	5 655
18·0	198·2	3 568	195·3	3 516	21·4	272	5 822	268	5 740
18·1	200·3	3 624	197	3 562	21·5	274·1	5 899	270·2	5 810

TABLE XVIII.—POWERS OF THE SPEED FOR SHIPS IN SALT WATER—*continued.*

V	$V^{1 \cdot 83}$	$V^{2 \cdot 83}$	$V^{1 \cdot 825}$	$V^{2 \cdot 825}$	V	$V^{1 \cdot 83}$	$V^{2 \cdot 83}$	$V^{1 \cdot 825}$	$V^{2 \cdot 825}$
21·6	276·7	5 977	272·5	5 890	25·0	361·8	9 040	355·8	8 890
21·7	279·4	6 056	275	5 970	25·1	364	9 143	358·5	9 000
21·75	280·3	6 095	276	6 000	25·2	366·6	9 246	361·1	9 100
21·8	281·3	6 135	277·1	6 045	25·25	368	9 298	362·6	9 150
21·9	284·1	6 215	279·3	6 110	25·3	369·6	9 350	364	9 190
22·0	286·2	6 296	281·7	6 199	25·4	372·5	9 455	366·4	9 310
22·1	289	6 377	284·1	6 290	25·5	375	9 561	369	9 410
22·2	291·2	6 459	286·2	6 360	25·6	377·6	9 668	371·8	9 510
22·25	292·1	6 500	287·6	6 400	25·7	380	9 775	374·4	9 630
22·3	293·8	6 542	288·8	6 448	25·75	382	9 829	375·9	9 680
22·4	296	6 625	291·1	6 520	25·8	383	9 883	377·1	9 740
22·5	298·1	6 709	293·6	6 600	25·9	385·2	9 992	379·9	9 840
22·6	300·8	6 794	296	6 690	26·0	388·6	10 101	382·2	9 940
22·7	303	6 880	298·3	6 780	26·1	392	10 212	385	10 060
22·75	304·6	6 923	299·6	6 805	26·2	394	10 323	387·7	10 150
22·8	306	6 966	300·7	6 860	26·25	395·2	10 379	389	10 210
22·9	308·1	7 053	303	6 940	26·3	397	10 435	390·5	10 280
23·0	310·3	7 140	305·6	7 028	26·4	399·5	10 547	393·2	10 380
23·1	313	7 228	308	7 120	26·5	402	10 661	396	10 490
23·2	315·7	7 317	310·4	7 200	26·6	405	10 775	398·7	10 610
23·25	316·9	7 362	311·9	7 250	26·7	407	10 890	401·4	10 710
23·3	318	7 407	313·1	7 300	26·75	409·8	10 948	402·9	10 770
23·4	320·1	7 497	315·5	7 390	26·8	411	11 006	404·1	10 820
23·5	323	7 588	318	7 480	26·9	413·6	11 123	406·7	10 940
23·6	325·2	7 680	320·3	7 560	27·0	416·5	11 240	409·4	11 050
23·7	328	7 772	323	7 660	27·1	419·2	11 358	412·4	11 180
23·75	329·6	7 818	324·1	7 700	27·2	421·5	11 477	415·1	11 300
23·8	330·8	7 865	325·4	7 750	27·25	423·5	11 537	416·5	11 370
23·9	332·5	7 959	327·8	7 830	27·3	425	11 597	418	11 410
24·0	335·9	8 054	330·2	7 926	27·4	428	11 718	420·7	11 520
24·1	338	8 149	332·7	8 015	27·5	431	11 839	423·5	11 630
24·2	341	8 245	335·2	8 110	27·6	433·6	11 961	426·3	11 770
24·25	342·1	8 296	336·5	8 160	27·7	435·5	12 084	429	11 890
24·3	343·3	8 342	337·8	8 205	27·75	438	12 146	430·4	11 960
24·4	346	8 440	340·2	8 300	27·8	440	12 208	431·8	12 010
24·5	348·2	8 538	343	8 405	27·9	442	12 333	434·6	12 120
24·6	350·7	8 637	345·6	8 500	28·0	445	12 458	437·4	12 250
24·7	353·3	8 737	348	8 600	28·1	447	12 585	440·2	12 390
24·75	355	8 787	349·4	8 650	28·2	451·8	12 712	443·3	12 500
24·8	356	8 837	350·6	8 700	28·25	451·7	12 776	444·8	12 570
24·9	359	8 938	353·2	8 800	28·3	454	12 840	446·2	12 630

TABLE XVIII.—POWERS OF THE SPEED FOR SHIPS IN SALT WATER—*continued.*

V	$V^{1\cdot83}$	$V^{2\cdot83}$	$V^{1\cdot825}$	$V^{2\cdot825}$	V	$V^{1\cdot83}$	$V^{2\cdot83}$	$V^{1\cdot825}$	$V^{2\cdot825}$
28·4	456·5	12 969	449	12 750	31·8	561	17 860	552·1	17 570
28·5	459·4	13 099	452	12 890	31·9	565	18 019	555	17 700
28·6	463	13 229	454·9	13 000	32·0	568	18 179	558·3	17 850
28·7	465	13 360	457·7	13 120	32·1	571·5	18 340	561·5	18 030
28·75	467·3	13 426	459·1	13 200	32·2	575	18 503	564·7	18 170
28·8	468·5	13 492	460·5	13 280	32·25	576	18 584	566·2	18 280
28·9	471·2	13 625	463·4	13 390	32·3	577·5	18 666	567·9	18 350
29·0	474·5	13 759	466·5	13 520	32·4	581	18 830	571	18 500
29·1	477	13 894	469·2	13 650	32·5	584·5	18 995	574·3	18 670
29·2	480·4	14 030	472·3	13 800	32·6	588·5	19 161	577·5	18 820
29·25	481·5	14 098	474	13 860	32·7	591	19 327	580·7	18 980
29·3	483·5	14 166	475·3	13 920	32·75	593	19 411	582·3	19 070
29·4	487	14 303	478·3	14 060	32·8	595	19 495	584	19 140
29·5	490	14 441	481·3	14 210	32·9	598	19 664	587	19 310
29·6	492	14 580	484·2	14 320	33·0	601	19 833	590·5	19 490
29·7	496	14 720	487·1	14 470	33·1	605	20 004	594	19 620
29·75	497·5	14 790	488·7	14 520	33·2	608·5	20 176	597	19 820
29·8	499	14 860	490	14 600	33·25	610	20 261	598·6	19 900
29·9	502·3	15 002	493·3	14 730	33·3	611	20 348	600	19 990
30·0	505	15 144	496·3	14 890	33·4	615	20 521	603·2	20 130
30·1	508	15 288	499·1	15 010	33·5	619	20 696	606·6	20 310
30·2	511	15 432	502·1	15 180	33·6	621·5	20 871	610	20 490
30·25	513	15 504	503·8	15 240	33·7	625	21 047	613·5	20 640
30·3	514	15 577	505·1	15 310	33·75	628·5	21 185	615	20 750
30·4	517·5	15 723	508·3	15 420	33·8	630	21 224	616·8	20 870
30·5	520·2	15 870	511·4	15 610	33·9	631·5	21 403	620	22 009
30·6	524	16 017	514·4	15 720	34·0	634·6	21 582	623·6	21 204
30·7	526	16 166	518	15 900	34·1	639	21 762	627	21 399
30·75	529	16 241	519·2	15 950	34·2	643	21 943	630·2	21 560
30·8	530	16 316	520·8	16 060	34·25	644·3	22 034	632	21 640
30·9	532·6	16 466	523·8	16 170	34·3	645	22 125	633·7	21 710
31·0	536	16 617	526·9	16 330	34·4	649	22 308	637	21 920
31·1	539	16 769	529·9	16 480	34·5	652	22 492	640·2	22 100
31·2	543	16 922	533	16 610	34·6	656	22 677	643·9	22 250
31·25	544	16 999	534·5	16 700	34·7	660	22 863	647·2	22 450
31·3	545·2	17 076	536·1	16 790	34·75	661	22 956	648·9	22 545
31·4	549·5	17 231	539·4	16 920	34·8	663	23 050	650·7	22 640
31·5	551·4	17 387	542·6	17 090	34·9	666	23 238	654·1	22 810
31·6	555	17 543	545·7	17 250	35·0	670	23 427	657·5	23 014
31·7	559	17 701	548·8	17 380	35·1	674	23 617	661	23 210
31·75	560	17 780	550·6	17 500	35·2	676	23 808	664·5	23 400

TABLE XVIII.—POWERS OF THE SPEED FOR SHIPS IN SALT WATER—*continued*.

V	$V^{1\cdot83}$	$V^{2\cdot83}$	$V^{1\cdot825}$	$V^{2\cdot825}$	V	$V^{1\cdot83}$	$V^{2\cdot83}$	$V^{1\cdot825}$	$V^{2\cdot825}$
35·25	678	23 904	666·2	23 480	38·7	804·5	31 133	790	30 580
35·3	680	24 000	668	23 560	38·75	808	31 247	791·8	30 690
35·4	683·5	24 192	671·3	23 770	38·8	809	31 361	793·7	30 800
35·5	687	24 386	674·9	23 940	38·9	811·5	31 591	797·3	31 000
35·6	690	24 581	678·2	24 130	39·0	816	31 821	801·1	31 250
35·7	694	24 777	682	24 370	39·1	820	32 052	804·8	31 480
35·75	696	24 875	683·5	24 460	39·2	824	32 285	808·7	31 700
35·8	698	24 974	685·2	24 550	39·25	826	32 402	810·5	31 810
35·9	701	25 172	688·8	24 710	39·3	828	32 519	812·4	31 930
36·0	705	25 371	692·2	24 920	39·4	831·5	32 753	816	32 160
36·1	709	25 571	695·7	25 120	39·5	835	32 989	820	32 400
36·2	712·5	25 772	699	25 300	39·6	840	33 226	823·7	32 600
36·25	714	25 873	701	25 400	39·7	843	33 464	827·5	32 880
36·3	716	25 974	702·8	25 500	39·75	845	33 583	829·4	33 000
36·4	719	26 177	706·1	25 710	39·8	847	33 703	831·2	33 120
36·5	723	26 381	709·8	25 920	39·9	850·3	33 943	835	33 330
36·6	726	26 586	713·2	26 120	40·0	854·5	34 185	839	33 560
36·7	729·5	26 792	716·9	26 220	40·1	859	34 427	851	34 200
36·75	732	26 895	718·6	26 380	40·2	863	34 670	851	34 210
36·8	734	26 999	720·3	26 530	40·25	864·5	34 792	852	34 300
36·9	738	27 207	724	26 700	40·3	867	34 915	854	34 400
37·0	741	27 417	727·7	26 925	40·4	871	35 161	857	34 610
37·1	745	27 627	731·5	27 150	40·5	874·5	35 408	860	34 810
37·2	748·5	27 838	735·2	27 390	40·6	878	35 656	863	35 020
37·25	750	27 944	737·1	27 480	40·7	883	35 905	865·7	35 270
37·3	753	28 050	739	27 570	40·75	884·5	36 030	867	35 350
37·4	756	28 364	743	27 790	40·8	886	36 155	869	35 410
37·5	760	28 478	746·7	28 000	40·9	891	36 406	872	35 650
37·6	764	28 693	750·4	28 210	41·0	894·5	36 659	875	35 840
37·7	768	28 910	754·4	28 420	41·1	898·5	36 912	878	36 080
37·75	769	29 018	756·3	28 530	41·2	901	37 167	881·3	36 330
37·8	770	29 127	758·4	28 650	41·25	904·5	37 295	883·2	36 410
37·9	774	29 346	762·1	28 890	41·3	907	37 423	884·6	36 500
38·0	779	29 566	764	29 033	41·4	911	37 680	888	36 720
38·1	782	29 786	768	29 230	41·5	914·5	37 938	892	37 020
38·2	786·5	30 008	771·3	29 450	41·6	918	38 197	896	37 290
38·25	788	30 119	773	29 580	41·7	922	38 458	901	37 600
38·3	790	30 231	775	29 700	41·75	924·5	38 588	903	37 700
38·4	793·5	30 455	778·8	29 880	41·8	928	38 719	905·5	37 810
38·5	797	30 680	782·3	30 150	41·9	931	38 982	909·5	38 050
38·6	801	30 906	786	30 350	42·0	934·5	39 246	914	38 370

TABLE XVIII.—POWERS OF THE SPEED FOR SHIPS IN SALT WATER—*continued*.

V	$V^{1.83}$	$V^{2.83}$	$V^{1.825}$	$V^{2.825}$	V	$V^{1.83}$	$V^{2.83}$	$V^{1.825}$	$V^{2.825}$
42·1	939	39 511	918	38 620	45·0	1 060	47 708	1 043	46 950
42·2	942	39 777	922	38 910	45·1	1 065	48 008	1 045	47 150
42·25	944·5	39 910	924	39 020	45·2	1 069	48 310	1 048·3	47 420
42·3	948·5	40 044	926	39 160	45·25	1 071	48 465	1 050·3	47 560
42·4	950·5	40 313	930·5	39 440	45·3	1 074	48 614	1 052·2	47 650
42·5	954·6	40 583	934·5	39 700	45·4	1 078	48 918	1 056·2	47 900
42·6	956	40 853	938·6	40 000	45·5	1 082	49 223	1 062·2	48 250
42·7	964	41 125	942·8	40 300	45·6	1 086	49 530	1 064·2	48 550
42·75	965	41 262	944·8	40 400	45·7	1 090	49 838	1 068	48 810
42·8	966·5	41 399	947	40 550	45·75	1 092	49 996	1 070·1	49 000
42·9	973	41 673	951	40 820	45·8	1 097	50 148	1 072·1	49 100
43·0	975	41 948	955·5	41 080	45·9	1 100	50 458	1 076	49 380
43·1	979·5	42 225	959·4	41 400	46·0	1 103	50 770	1 080	49 650
43·2	984	42 503	963·6	41 650	46·1	1 109	51 083	1 084	50 000
43·25	986	42 642	966	41 820	46·2	1 112	51 397	1 088	50 300
43·3	988	42 782	967·8	41 900	46·25	1 116	51 558	1 090	50 450
43·4	993	43 062	972	42 150	46·3	1 119	51 712	1 092	50 600
43·5	996	43 343	976	42 460	46·4	1 123	52 029	1 096	50 850
43·6	1 000	43 626	980	42 780	46·5	1 127	52 347	1 100	51 100
43·7	1 004	43 910	984	43 000	46·6	1 131	52 666	1 104	51 500
43·75	1 008	44 052	986·5	43 130	46·7	1 135	52 986	1 108	51 800
43·8	1 110	44 195	988·8	43 300	46·75	1 138	53 151	1 110	51 900
43·9	1 013	44 481	993	43 560	46·8	1 140	53 308	1 113	52 110
44·0	1 018	44 768	998	43 900	46·9	1 145	53 632	1 116	52 320
44·1	1 021	45 057	1 002	44 300	47·0	1 149	53 956	1 121	52 700
44·2	1 025	45 347	1 006·3	44 500	47·1	1 152	54 281	1 125·7	53 010
44·25	1 028	45 492	1 007·5	44 600	47·2	1 158	54 608	1 131	53 450
44·3	1 030	45 638	1 011	44 810	47·25	1 160	54 776	1 133	53 550
44·4	1 035	45 930	1 015·2	45 060	47·3	1 162	54 936	1 136	53 620
44·5	1 039	46 223	1 020	45 400	47·4	1 167	55 265	1 140·5	54 100
44·6	1 044	46 518	1 024·3	45 750	47·5	1 171	55 596	1 145	54 400
44·7	1 047	46 814	1 029	46 000	47·6	1 175	55 928	1 150	54 800
44·75	1 050	46 962	1 031·6	46 190	47·7	1 179	56 261	1 155	55 100
44·8	1 054	47 110	1 034	46 330	47·75	1 181	56 432	1 158	55 300
44·9	1 057	47 408	1 039	46 600	47·8	1 184	56 595	1 160	55 450

The rate of increase of horse-power for small increments of speed may be ascertained by the use of common logarithms. We have I.H.P. $\sim V^n$.

Take, for example, the two highest speeds of the Dutch tugboat:

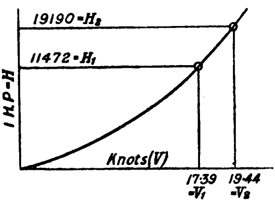
V_1	10·84 knots at 230·58 I.H.P.	I.H.P.$_1$
V_2	11·01 knots at 260·32 I.H.P.	I.H.P.$_2$

$$\text{Log } V_2 \div V_1 = \log 11{\cdot}01 - \log 10{\cdot}84 = {\cdot}006\,69.$$
$$\text{Log I.H.P.}_2 \div \text{I.H.P.}_1 = \log 260{\cdot}32 - \log 230{\cdot}58 = {\cdot}053\,62.$$

By dividing ·053 62 by ·006 69 we obtain n, the index of the power of V, according to which the I.H.P. varies. In this case $n = 8{\cdot}01$.

Taking the speeds 9·02 and 10·07 knots, $n = 4{\cdot}53$.

In the preparation of the first edition of *Steamship Coefficients, Speeds and Powers*, n was found graphically, by measuring the tangent of the angle of slope of the curve,—a laborious process compared with the logarithmic method.



$$H \sim V^n$$
$$H = a V^n$$
$$\log H = \log a + n \log V$$
$$\frac{H_2}{H_1} = \left(\frac{V_2}{V_1}\right)^n$$
$$\log (H_2 \div H_1) = n \log (V_2 \div V_1)$$
$$\log H_2 - \log H_1 = n\,[\log V_2 - \log V_1]$$

$$
\begin{aligned}
4{\cdot}283\,075\,0 &= \log H_2 \\
4{\cdot}059\,639\,1 &= \log H_1 \\
\hline
0{\cdot}223\,435\,9 &= \log H_2 - \log H_1
\end{aligned}
$$

$$\therefore n = \frac{\log H_2 - \log H_1}{\log V_2 - \log V_1}$$

$$
\begin{aligned}
1{\cdot}288\,696\,3 &= \log V_2 \\
1{\cdot}240\,299\,6 &= \log V_1 \\
\hline
0{\cdot}048\,396\,7 &= \log V_2 - \log V_1
\end{aligned}
$$

$$n = \frac{{\cdot}223\,435\,9}{{\cdot}048\,396\,7}$$

$$\log n = \log {\cdot}223\,435\,9 - \log {\cdot}048\,396\,7$$

$$
\begin{aligned}
\log n &= \bar{1}{\cdot}349\,153\,2 - (\bar{2}{\cdot}684\,815\,8) \\
&= -1 + {\cdot}349\,153\,2 - (-2 + {\cdot}684\,815\,8) \\
&= -1 + {\cdot}349\,153\,2 + 2 - {\cdot}684\,815\,8 \\
&= 1 + {\cdot}349\,153\,2 - {\cdot}684\,815\,8 \\
&= {\cdot}664\,337\,4 \\
&= \log 4{\cdot}616\,7
\end{aligned}
$$

$$\therefore n = 4{\cdot}616\,7.$$

$$
\begin{array}{r}
1{\cdot}349\,153\,2 \\
{\cdot}684\,815\,8 \\
\hline
{\cdot}664\,337\,4
\end{array}
\qquad
\begin{array}{r}
\bar{1}{\cdot}349\,153\,2 \\
\text{or } \bar{2}{\cdot}684\,815\,8 \\
\hline
{\cdot}664\,337\,4
\end{array}
$$

Definitions.—Entrance and run = that portion of the bow and stem respectively which is clear of the perfectly parallel midship body. In any given ship, as the draught is reduced, the entrance and run become finer. This should be remembered when calculating wake values for propeller design. The word *form* applies to the shape of a ship, apart from dimensions and proportions.

Limiting Economical Speed.

In a paper read before the Institution of Engineers and Shipbuilders in Scotland in 1910, Mr P. A. Hillhouse, B.Sc., showed an empirical relation between block coefficient, length, and limit of economical speed.

If L = length of ship,
M = length of parallel middle body,
E = combined length of curved ends,
b = block coefficient,
m = midship section coefficient,
= block coefficient of parallel middle body,
e = block coefficient of ends,
$\frac{b}{m}$ = prismatic coefficient = p,
$\frac{e}{m}$ = prismatic coefficient of ends = p,
then E+M = L.

Supposing the end lengths, E, to be divided into a number of uniformly spaced sections, "and any desired increase of block coefficient obtained by shrinking up the ends, so that the said sections would be closer together, but still of the same shapes and uniformly spaced, the parallel body being lengthened to fill up the gaps so formed." For a series of vessels of block coefficient between about ·56 and ·78, and at speeds proportional to the square root of the combined length of the ends, Mr Hillhouse found that the Admiralty coefficient $\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$ was practically constant, and at speeds above about ·925 $\sqrt{E}$ wave-making rapidly increased.

$$V = \cdot 925\sqrt{E}$$
$$= \cdot 925\sqrt{[2\cdot 563\,(1-p)L]}$$
$$\frac{V}{\sqrt{L}} = 1\cdot 482\,\sqrt{(1-p)}$$

for smooth water trials on measured course

$$= \cdot 52\,\frac{\sqrt{1-p}}{\cdot 35}.$$

Values of $\dfrac{V}{\sqrt{L}}$ are illustrated in Mr Hillhouse's table below.

Plate 13 shows average practice under moderate weather conditions at sea.

TABLE XIX.

b.	*m*.	*p*.	Percentage M.	Percentage E.	$\frac{V}{\sqrt{L}}$ Smooth water trials.	Sea speeds $\frac{V}{\sqrt{L}}$ Roughly about ·91 trial speed.
·560	·918	·61	0	100	·925	·842
·582	·924	·63	5·13	94·87	·897	·816
·604	·929	·65	10·26	89·74	·869	·790
·626	·934	·67	15·38	84·62	·842	·767
·648	·939	·69	20·51	79·49	·814	·741
·670	·944	·71	25·64	74·36	·786	·715
·692	·948	·73	30·77	69·23	·758	·690
·714	·952	·75	35·90	64·10	·730	·665
·736	·956	·77	41·02	59·98	·703	·640
·758	·960	·79	46·15	53·85	·675	·614
·780	·963	·81	51·28	48·72	·647	·589

Data from Mr John Neill's remarks in the discussion on Mr E. Saxton White's paper read before the North-East Coast Institution of Engineers and Shipbuilders, session 1911–1912.

No.	Length.	Beam.	Draught.	Displacement.	Coefficients. Block.	Coefficients. Prism.	Coefficients. Mid area.	Speed in knots.	Residuary horse-power. Taylor's standard tank data.	Residuary horse-power. Actual tank results.
1	405	54·8	24·4	10 000	·648	·70	·926	16·1	1 512	
1*a*	405	70·6	18·9	10 000	·648	·70	·926	16·1	1 675	
2	449	55·3	24·6	10 000	·574	·62	·926	21·2	5 070	
2*a*	449	71·3	19·0	10 000	·574	·62	·926	21·2	5 300	
3	500	55·2	24·5	10 000	·518	·56	·926	25·7	7 110	
3*a*	500	71·2	19·0	10 000	·518	·56	·926	25·7	8 240	
4	404	55·5	22·8	10 000	·684	·71	·962	15·85	1 505	1 570
5	428	66·6	22·5	10 000	·539	·61	·884	22·1	7 210	7 690
6	405	67·6	21·25	10 000	·600	·613	·98	19·1	3 230	2 920

The forms 1*a*, 2*a*, and 3*a* only differ from 1, 2, and 3 respectively in having a differenl ratio of beam to draught.

The forms 4, 5, and 6 were actual ships which had been built and tried.

Data from paper by Mr Ernest Saxton White, B.Sc., read before the North-East Coast Institution of Engineers and Shipbuilders, session 1911–1912.

Tons displacement.	Type of ship.	Length on water-line.	Beam over shell.	Draught ex hanging keel.	Block coefficient.	Prismatic coefficient.	Midship section coefficient.	Maximum speeds.	I.H.P.	$\frac{\Delta V^3}{I.H.P.}$
		feet	feet	feet						
10 090	S.S.	374	49·5	24·0	·794	·814	·977	11·2	2 620	250
10 140	S.S.	420	55·0	19·78	·777	·799	·973	12·54	3 530	262
10 190	T.S.S.	449	55·1	20·25	·712	·738	·965	16·0	7 290	265
10 120	T.S.S.	428	55·0	24·2	·623	·677	·92	17·05	9 600	242
10 190	T.S.S.	481	56·6	21·7	·602	·646	·932	19·9	14 900	249
10 200	T.S S.	432	66·0	23·3	·537	·59	·91	22·1	19 050	267
10 090	T.S.S.	500	62·2	23·3	·488	·556	·877	26·1	34 600	240
10 150	T.S.S.	506	60·3	23·3	·50	·569	·88	25·92	32 300	253
10 020	T.S.S.	485	63·15	23·4	·49	·555	·884	25·7	32 600	242

Vessels of 10 000 tons displacement. Data from Mr Hinchliffe's remarks in the discussion on above paper.

Type of vessel.	Length of water-line.	Beam.	Mean draught.	Coefficients.			Speed in knots.	Residuary resistance E.H.P. from model.	Authority.	Residuary resistance in lbs. per ton of displacement.
				Block.	Midship.	Prismatic.				
Torpedo boat destroyer	588·1	61·42	19·21	·504 4	·76	·664	40·4	79 300	Mr A. W. Johns	64
Scout	583·2	60·43	21·53	·461 3	·88	·525	30·7	18 360	,,	19·45
Cruiser	455·7	67·25	24 02	·475 5	·887	·536	22·0	3 540	Mr R. E. Froude	5·25
Battleship	365·9	65·81	24·13	·602 4	·905	·665	17·8	4 662	Mr A. W. Johns	8·46
Merchant steamer	418·4	52·3	24·41	·655 3	·963 8	·677 8	17·39	1 703	Prof. H. C. Sadler	3·33
,,	403·5	50·4	23·5	733	·964	·760	14·05	885	,,	2·05
,,	382·9	47·9	22·33	·855	·984	·869	9·8	382	,,	1·27

The following approximate formula, based upon some investigations by Hovgaard, may be found useful for roughly determining length appropriate to speed, taking account of the transverse bow-waves,

$$\frac{(\text{Speed in knots})^2}{1{\cdot}8} = n.$$

The denominator given here as 1·8 for average intermediate merchant ships seems to vary slightly according to the angle of entrance,—*i.e.* 1·8 corresponds to a certain mean angle frequently found.

Then, if $\frac{\text{Length of ship in feet}}{n}$ is a whole number, the length is unsuitable.

If $\frac{\text{Length of ship in feet}}{n}$ is, say, 4·5 or 4·4, or 3·5, 3·7, 5·6, or 4·6, or other number representing a hollow between wave-crest of the wave system formed by one end of the ship, and crest of any transverse wave formed by the other end of the ship, then the length is favourable.

The question of absolute size in relation to speed is difficult. In Mr J. J. O'Neill's elaborate and suggestive paper to the Institution of Engineers and Shipbuilders in Scotland, 1907–8, entitled "The Interrelation of Theory and Practice in Shipbuilding," curves are given showing, for certain types of large fast mail steamers, the effect, upon the limits of economical speed, of developing dimensions. This should be read after a study of Mr R. E. Froude's 1904 paper to the Institution of Naval Architects (see p. 75). Research work has only begun on this point in its bearing upon ordinary cargo and passenger ships. It is important to every shipowner, when laying down a new vessel, to know if it is of a length favourable for the intended speed.

Mr Hillhouse gives the following table for a relation between prismatic coefficient and speed-length ratio (trial trip speeds).

Prismatic coefficient.	Speed-length ratio.	Prismatic coefficient.	Speed-length ratio.
·61	·925	·73	·758
·63	·897	·75	·730
·65	·869	·77	·703
·67	·842	·79	·675
·69	·814	·81	·647
·71	·786		

The following table by Mr Hillhouse shows the value of the Admiralty coefficient $\left(\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}\right)$ for various lengths of ships on trial, assuming propulsive coefficient = ·55.

TABLE XX.

Length on water-line.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Length on water-line.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
100	137	500	307
150	188	550	310
200	227	600	312
250	255	650	315
300	275	700	317
350	289	750	319
400	297	800	321
450	303	850	323

Most estimators have their own private curves or tables, broadly indicating the relation of block coefficient to speed-length ratio, for a given class of vessel. Plates 14, 17, 39 and Table XXI illustrate something of this kind, and are intended as a rough guide for average results in ordinary weather under moderately good steaming conditions. Methodical proportioning of vessel, with due regard to absolute size and shape of transverse sections, may produce results better than those indicated by the curves.

[TABLE

TABLE XXI.—FINENESS APPROPRIATE TO SPEED ON SERVICE UNDER AVERAGE GOOD CONDITIONS.

Coefficients.			$\frac{V}{\sqrt{L}}$.	Speed in knots for ships of various lengths.					
Block.	Prismatic.	Mid area.		50 ft.	100 ft.	150 ft.	200 ft.	250 ft.	300 ft.
·85	·86	·988	·43	3·04	4·3	5·26	6·09	6·8	7·45
·82	·833	·985	·48	3·395	4·8	5·88	6·79	7·59	8·31
·80	·814	·984	·513	3·625	5·13	6·29	7·25	8·1	8·89
·77	·785	·981	·566	4·0	5·66	6·94	8·0	8·95	9·8
·76	·775	·980	·584	4·13	5·84	7·15	8·25	9·23	10·1
·75	·767	·979	·60	4·245	6·0	7·35	8·49	9·49	10·4
·74	·757	·977	·62	4·39	6·2	7·6	8·77	9·8	10·73
·72	·739	·975	·655	4·64	6·55	8·02	9·25	10·36	11·36
·70	·721	·971	·692	4·9	6·92	8·47	9·79	10·93	11·99
·68	·703	·968	·73	5·16	7·3	8·94	10·32	11·53	12·64
·67	·694	·966	·749	5·3	7·49	9·16	10·59	11·82	12·97
·65	·677	·961	·79	5·59	7·9	9·68	11·17	12·49	13·69
·645	·673	·960	·80	5·66	8·0	9·8	11·31	12·65	13·86
·63	·66	·955	·832	5·88	8·32	10·19	11·76	13·15	14·4
·62	·651	·952	·856	6·05	8·56	10·5	12·1	13·54	14·83
·61	·641	·951	·88	6·22	8·8	10·78	12·45	13·9	15·22
·60	·634	·947	·905	6·4	9·05	11·09	12·8	14·3	15·68
·58	·615	·943	·957	6·76	9·57	11·72	13·53	15·12	16·58
·55	·589	·935	1·04	7·35	10·4	12·72	14·7	16·42	18·0
·53	·57	·930	1·105	7·8	11·05	13·54	15·6	17·45	19·11
·52	·561	·927	1·14	8·05	11·4	13·98	16·1	18·0	19·71
·51	·554	·921	1·178	8·31	11·78	14·4	16·62	18·6	20·39
·50	·549	·912	1·217	8·6	12·17	14·9	17·2	19·22	21·04
·49	·543	·904	1·254	8·86	12·54	15·37	17·71	19·81	21·71
·48	·540	·889	1·297	9·16	12·97	15·88	18·32	20·49	22·42
·47	·539	·873	1·342	9·5	13·42	16·43	19·0	21·21	23·22
·46	·537	·856 5	1·391	9·84	13·91	17·05	19·68	22·0	24·1
·45	·538	·837	1·444	10·21	14·44	17·7	20·4	22·81	25·0
·44	·540	·815	1·498	10·6	14·98	18·32	21·18	23·62	25·9
·43	·543	·793	1·56	11·04	15·6	19·1	22·02	24·66	27·0
·42	·550	·764	1·623	11·48	16·23	19·88	22·92	25·65	28·11
·41	·565	·726	1·69	11·96	16·9	20·69	23·9	26·7	29·23
·40	·587	·682	1·766	12·5	17·66	21·61	24·98	27·9	30·6

TABLE XXI.—FINENESS APPROPRIATE TO SPEED ON SERVICE UNDER AVERAGE GOOD CONDITIONS—*continued.*

Coefficients.			$\frac{V}{\sqrt{L}}$.	Speed in knots for ships of various lengths.					
Block.	Prismatic.	Mid area.		350 ft.	400 ft.	450 ft.	500 ft.	550 ft.	600 ft.
·85	·86	·988	·43	8·04	8·6	9·11	9·61	10·09	10·53
·82	·833	·985	·48	8·97	9·6	10·19	10·73	11·28	11·76
·80	·814	·984	·513	9·59	10·24	10·88	11·46	12·01	12·55
·77	·785	·981	·566	10·6	11·33	12·0	12·67	13·29	13·88
·76	·775	·980	·584	10·91	11·68	12·39	13·04	13·69	14·3
·75	·767	·979	·60	11·21	12·0	12·72	13·41	14·08	14·7
·74	·757	·977	·62	11·6	12·4	13·16	13·87	14·54	15·2
·72	·739	·975	·655	12·24	13·1	13·9	14·65	15·37	16·05
·70	·721	·971	·692	12·92	13·82	14·68	15·47	16·21	16·93
·68	·703	·968	·73	13·65	14·6	15·49	16·31	17·12	17·88
·67	·694	·966	·749	14·0	14·99	15·89	16·72	17·56	18·32
·65	·677	·961	·79	14·78	15·8	16·75	17·66	18·51	19·35
·645	·673	·960	·80	14·97	16·0	16·98	17·89	18·76	19·6
·63	·66	·955	·832	15·56	16·61	17·63	18·6	19·5	20·39
·62	·651	·952	·856	16·0	17·12	18·18	19·16	20·1	21·0
·61	·641	·951	·88	16·46	17·6	18·67	19·69	20·62	21·56
·60	·634	·947	·905	16·91	18·1	19·2	20·22	21·21	22·19
·58	·615	·943	·957	17·9	19·14	20·3	21·4	22·42	23·43
·55	·589	·935	1·04	19·44	20·8	22·07	23·25	24·4	25·48
·53	·57	·930	1·105	20·63	22·1	23·41	24·71	25·9	27·08
·52	·561	·927	1·14	21·31	22·8	24·2	25·5	26·75	27·92
·51	·554	·921	1·178	22·0	23·53	24·98	26·32	27·6	28·81
·50	·549	·912	1·217	22·75	24·35	25·8	27·2	28·55	29·8
·49	·543	·904	1·254	23·43	25·06	26·6	28·02	29·4	30·7
·48	·540	·889	1·297	24·23	25·92	27·49	29·0	30·4	31·76
·47	·539	·873	1·342	25·1	26·82	28·5	30·0	31·5	32·9
46	·537	·856 5	1·391	26·02	27·81	29·55	31·15	32·64	34·1
·45	·538	·837	1·444	27·0	28·88	30·61	32·29	33·85	35·38
·44	·540	·815	1·498	28·0	29·96	31·75	33·43	35·09	36·65
·43	·543	·793	1·56	29·2	31·2	33·1	34·88	36·6	38·2
·42	·550	·764	1·623	30·38	32·43	34·4	36·3	38·06	39·79
·41	·565	·726	1·69	31·6	33·8	35·8	37·8	39·62	41·4
·40	·587	·682	1·766	33·02	35·3	37·43	39·5	41·45	43·25

TABLE XXI.—FINENESS APPROPRIATE TO SPEED ON SERVICE UNDER AVERAGE GOOD CONDITIONS—*continued.*

Coefficients.			$\frac{V}{\sqrt{L}}$	Speed in knots for ships of various lengths.					
Block.	Prismatic.	Mid area.		650 ft.	700 ft.	750 ft.	800 ft.	850 ft.	900 ft.
·85	·86	·988	·43	10·96	11·39	11·78	12·17	12·53	12·9
·82	·833	·985	·48	12·22	12·7	13·14	13·59	14·0	14·4
·80	·814	·984	·513	13·09	13·58	14·03	14·5	14·92	15·38
·77	·785	·981	·566	14·42	14·98	15·5	16·0	16·5	17·0
·76	·775	·980	·584	14·89	15·42	15·99	16·5	17·0	17·5
·75	·767	·979	·60	15·3	15·88	16·41	16·97	17·49	18·0
·74	·757	·977	·62	15·8	16·4	17·0	17·51	18·09	18·6
·72	·739	·975	·655	16·7	17·31	17·92	18·51	19·1	19·65
·70	·721	·971	·692	17·64	18·3	18·95	19·58	20·19	20·76
·68	·703	·968	·73	18·61	19·3	20·0	20·65	21·29	21·89
·67	·694	·966	·749	19·1	19·8	20·5	21·19	21·8	22·42
·65	·677	·961	·79	20·15	20·9	21·61	22·33	23·0	23·69
·645	·673	·960	·80	20·4	21·19	21·9	22·61	23·32	24·0
·63	·66	·955	·832	21·2	22·0	22·79	23·54	24·23	24·95
·62	·651	·952	·856	21·81	22·62	23·42	24·21	24·96	25·69
·61	·641	·951	·88	22·41	23·28	24·11	24·88	25·62	26·4
·60	·634	·947	·905	23·05	23·95	24·79	25·6	26·39	27·16
·58	·615	·943	·957	24·4	25·31	26·2	27·04	27·88	28·7
·55	·589	·935	1·04	26·51	27·5	28·46	29·4	30·3	31·2
·53	·57	·930	1·105	28·18	29·2	30·22	31·23	32·21	33·18
·52	·561	·927	1·14	29·1	30·19	31·21	32·25	33·21	34·2
·51	·554	·921	1·178	30·0	31·15	32·24	33·3	34·3	35·3
·50	·549	·912	1·217	31·0	32·2	33·3	34·4	35·44	36·5
·49	·543	·904	1·254	31·97	33·19	34·35	35·45	36·54	37·6
·48	·540	·889	1·297	33·07	34·3	35·5	36·65	37·8	38·88
·47	·539	·873	1·342	34·22	35·52	36·79	38·0	39·16	40·3
·46	·537	·856 5	1·391	35·5	36·81	38·14	39·38	40·6	41·75
·45	·538	·837	1·444	36·8	38·2	39·55	40·85	42·1	43·3
·44	·540	·815	1·498	38·16	39·6	41	42·35	43·6	44·9
·43	·543	·793	1·56	39·8	41·3	42·68	44·1	45·5	46·8
·42	·550	·764	1·623	41·4	42·95	44·45	45·9	47·4	48·65
·41	·565	·726	1·69	43·1	44·7	46·3	47·8	49·3	50·6

In the Channel steamers "Normannia" and "Hantonia" the beam was small (36 ft.), the stability being obtained by filling out the water-line aft and lengthening the parallel line of the water plane aft to get more length, upon which moment of inertia could be obtained, in order to produce the same B.M. as a broader ship. Instead of having 39-ft. beam as in "Cæsarea" and "Sarnia," the beam was made 3 ft. less, *i.e.* "the area was taken off amidships and put on the after end of the ship." The effect upon the resistance of that change was exactly in accordance with what Dr W. Froude pointed out a generation previously, viz. that if the water-line forward is kept no fuller, and if the beam is not increased, the water-line may be varied with impunity, provided the cross-sectional areas are kept the same. Dr W. Froude showed, in fact, broadly speaking, that resistance depended on the beam of the ship, the curve of cross-sectional areas, and the fineness of the surface water-line forward. The designers of the "Normannia" and "Hantonia" chose such a water-line with the reduced beam as would give a sufficient moment of inertia to produce the same B.M. as a broader ship.

Experiments made with models in artificial waves at Messrs Denny's tank, Dumbarton, indicate that even in full ships different forms of the fore body have a marked influence on the resistance amongst waves, but from model experiments we can only estimate the probable performances of ships of different fulnesses. Only experience with ships at sea can show whether ·77 block coefficient, say, is much more adversely affected by rough weather than a finer block. Recent experience has shown cargo vessels of ·74 to ·75 more capable of maintaining regularity of service than steamers of fuller block, but the amount of flare and the best sections of under-water fore body are still moot points.

The following is tabulated from information given in an article in *The Engineer*, Feb. 4, 1916 :—

	Fast liners.	Full cargo vessels.
Combined influence of waves and a following wind of 50 knots.	Causes decrease of speed of 3 per cent.	Causes decrease of speed of 11 per cent.
Strong fair wind and a following sea.	Loss of speed or equivalent coal consumed 10 per cent.	Loss of speed or equivalent coal consumed 40 per cent.

	Fast liners.	Full cargo vessels.
Head wind of 30 knots with accompanying sea.	Decrease of speed 2 per cent. Increase of power 6 per cent.	Decrease of speed 9 per cent. Increase of power 30 per cent.
Head wind of 50 knots (heavy gale) with head sea.	Reduction of speed 25 per cent. Power 100 per cent. more than for the same speed in smooth water.	Reduction of speed 64 per cent. Power 300 per cent. more than for the same speed in smooth water.

Both for smooth-water conditions and for rough water, especially in full cargo ships, the U-shaped upright forward sections and V-shaped sections aft are approved. Rear-Admiral Taylor says: "Pitching exaggerates nearly all causes of speed loss. If it were possible to devise a vessel which would not pitch, she would lose much less speed in rough water than one that does pitch." Regarding the features which minimise pitching, "the preponderance of opinion is probably in favour of the U-shaped bow type and rather full-bow water-lines." (Probably this form is beneficial both in waves and in smooth water.)

CHAPTER VII.

APPLICATION OF TAYLOR'S CONTOURS FOR RESIDUARY RESISTANCE PER TON Δ.

BEFORE using these to predict the resistance of a merchant-ship type whose dimensions and features of form are known, we must apply certain corrections to bring the two into line.

(1) We must remember that Taylor's standard series has a cruiser stern, and that Taylor's length is l.w.l. His upper water-lines therefore have an advantage over those of the stern of an ordinary merchant ship in being carried further aft. Taylor's ship must be first considered shortened at the stern, by a proportion of the length of the immersed counter determined by judgment. The ratio of length to beam must be reduced, and the block coefficient, prismatic coefficient, and displacement-length ratio increased.

For the same reason, when comparing the results of Mr R. E. Froude's experiments with those of Mr D. W. Taylor, we must make a similar correction, remembering that Froude's length is length b.p., while Taylor's length is l.w.l.,—though both have the cruiser stern. Froude's vessels therefore have an advantage. The opposite is found when we come to Mr Baker's 1913 models and Professor Sadler's 1907–1909 types, which are mercantile ship forms, where the aft perpendicular is the end of the water-line; therefore, before using Taylor's contours of resistance, fuller and shorter forms must be taken than those corresponding to the dimensions of the merchant ships in question. In using Froude's results to compare with those of Taylor, Froude's length (*i.e.* length b.p.) should first be modified by lengthening, *i.e.* correcting it to what is more nearly a water-line length.

In Mr R. E. Froude's 1904 models, displacement includes immersed counter and ram; length for prismatic and block coefficients is measured from midship section to perpendiculars; draught is that at midship section.

In shortening Type 1 to obtain Type 4, the length of the aft

body measured to the rudder post was shortened 20 ft., but as the water-line overhang was increased, the actual water-line shortening was less than the nominal 20 ft.

In Mr Wall's paper to the Liverpool Engineering Society in 1915, the estimated advantage due to the increased water-line length of the cruiser stern as compared with the ordinary type of stern, is worked out as giving a channel steamer of 350 ft. length b.p., an increased water-line length of 363 ft. 3 ins., and a consequent gain of $\frac{4}{10}$ths of a knot in speed (and, with the possible reduction of beam, half a knot increase in speed).

(2) The value of the ratio $\frac{\text{Beam}}{\text{Draught}}$ to be used with Taylor's contours must be modified to correspond with his midship section coefficient, ·926. So long as we keep the (draught × midship-section coefficient) constant, we may alter the draught with impunity. This follows from Mr Froude's dictum in his paper to the Institution of Naval Architects in 1904, viz.: "We may almost say that the resistance of a form is determined solely by *the curve of cross-section areas, together with the extreme beam and the surface water-line of the fore body*; and if these are adhered to, the lines may be varied in almost any reasonable way without materially increasing or decreasing the resistance at any speed."

That is to say, ships of the same *length, beam, area of midship section, surface fore-body water-line,* and the same *curve of cross-sectional areas,* will have approximately the same resistance at any given speed. The prismatic coefficient and the value of $\frac{\Delta}{\left(\frac{L}{100}\right)^3}$, taken together, determine the area of midship section. The beam is equal to $\frac{\text{Midship area}}{\text{Mean depth section}}$ or $\frac{\text{Midship area}}{\text{Draught} \times \text{coefficient.}}$

Before comparing results of ships in which $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}25$, and midship area coefficient = ·98, with Taylor's contours (based upon his standard midship-area coefficient of ·926), the beam-draught ratio must be altered to what it would be if the ship in question had a midship-section coefficient of ·926, the new beam-draught ratio being $\frac{B_1}{H_1} = \frac{B}{H} \times \frac{\cdot 926}{\cdot 98} = 2{\cdot}129$.

Plate 10 shows profile and part of curve of sectional areas of Mr Baker's Set A, 1913. As in Taylor's plans, the stations are drawn at intervals of one-twentieth of the ship's length. Measuring the ordinates of the stern end of the curve, we find that the

last station scales ·063 of the midship ordinate, and the penultimate ordinate = ·175 of the midship ordinate. The corresponding ordinates of Taylor's standard series, for the same prismatic coefficient, are respectively about ·085 and ·208 of his midship ordinate, showing that, approximately for the same prismatic coefficient, Mr Baker's stern lines are finer than Mr Taylor's. As the shortening and sharpening of Mr Baker's lines is probably almost entirely accounted for by the fact that his model has about 10 per cent. of parallel body, and that the stern lines would perhaps be almost identical with Mr Taylor's, if Mr Baker's, like Mr Taylor's, had no parallel body, there is not likely to be much error in comparing the results of Mr Baker's ships of a given l.b.p. with Mr Taylor's standard series direct, without applying any correction to the length.

In correcting Mr R. E. Froude's 1904 Series A, for comparison with Taylor, the overhang of the stern of the Froude ship should be added to the length b.p., to obtain the length of ship to which Taylor's residuary resistance contours apply, thus :—

TABLE XXII.

Froude Type No.	Froude's l.b.p. in feet.		Overhang in feet.	Taylor's length.		Froude's length b.p. / Taylor's length	
Type 1. .	350		13	363		·965	·96
,, 2. .	340		15	355		·959	
,, 3. .	330	320	15·5	345·5	335·5	...	
,, 4. .	325			340·5		·954	
,, 5. .	320			335·5			
,, 6. .	310			325·5			

The beam-draught ratio of Mr R. E. Froude's 1904 models, Series A, would similarly be multiplied by $\dfrac{\cdot 926}{\cdot 877\,5}$.

(3) A third correction to apply to our vessel before applying Taylor's contours is that of eliminating the effect of parallel body, which is absent in Taylor's standard series. Figs. 126–129, in Mr Taylor's book, "*The Speed and Power of Ships*, vol. ii, Plates," give a guide to the direction in which various percentages of parallel body influence the resistance for different speed-length ratios and prismatic coefficients—sometimes increasing, sometimes decreasing, the resistance. Our Plate 11 shows practicable per-

centages of length of ship to which parallel body should be given for various speed-length ratios.

Taylor's 1913 models. 500-ft. ship. $\Delta = 17\,850$ tons. Block coefficient = ·60. Mid-area coefficient = ·92. Prismatic = ·652 2.

$$\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 142{\cdot}9. \quad \frac{B}{H} = 2{\cdot}4. \quad \frac{V}{\sqrt{L}} = {\cdot}895.$$

From the deep-water resistance curves we find that

At 20 knots,	E.H.P. =	11 030
	Skin H.P. =	6 050
	Residuary H.P. =	4 980

$$\left.\begin{array}{c}\text{Residuary resist-}\\\text{ance in lbs.}\end{array}\right\} = \frac{\text{Residuary H.P.}}{\text{Speed in knots} \times {\cdot}003\,07} = \frac{4\,980}{20 \times {\cdot}003\,07} = 81\,100.$$

$$\left.\begin{array}{c}\text{Residuary resistance in lbs.}\\\text{per ton of displacement}\end{array}\right\} = \frac{81\,100}{17\,850} = 4{\cdot}55.$$

At 18 knots, $\dfrac{V}{\sqrt{L}} = {\cdot}806.$

7 150 E.H.P.
4 500 skin H.P.

2 650 residuary H.P.

2·682 lbs. residuary resistance per ton Δ,

about 19 per cent. higher than that of Taylor's standard series at speed-length-ratio of ·806.

Let us find the residuary resistance in lbs. per ton Δ from Taylor's standard contours :—

	$\frac{V}{\sqrt{L}}$.	Residuary resistance in lbs. per ton Δ corresponding to values of $\frac{B}{H}$.		
		2·25.	3·75.	2·415.
20 knots	·850	2·825	3·473	2·896 3
	·900	4·264	5·000	4·345
	·895	...	...	4·192 8
18 knots	·850	2·825	3·473	2·896 3
	·800	2·132	2·50	2·172 5
	·806	...	...	2·259 4

New value of $\frac{B}{H} = \frac{B_1}{H_1} = 2{\cdot}4 \times \frac{{\cdot}926}{{\cdot}92} = 2{\cdot}415$

$$\begin{array}{r} 3{\cdot}75 \\ 2{\cdot}25 \\ \hline 1{\cdot}50 \end{array} \qquad \begin{array}{r} 2{\cdot}415 \\ 2{\cdot}250 \\ \hline {\cdot}165 \end{array} \qquad \begin{array}{r} 3{\cdot}473 \\ 2{\cdot}825 \\ \hline {\cdot}648 \end{array} \qquad \begin{array}{r} 2{\cdot}825 \\ {\cdot}071\,3 \\ \hline 2{\cdot}896\,3 \end{array}$$

$$\begin{array}{r} {\cdot}895 \\ {\cdot}850 \\ \hline {\cdot}045 \end{array} \qquad \begin{array}{r} {\cdot}900 \\ {\cdot}850 \\ \hline {\cdot}050 \end{array} \qquad \begin{array}{r} 5{\cdot}000 \\ 4{\cdot}264 \\ \hline {\cdot}736 \end{array} \qquad \begin{array}{r} 4{\cdot}264 \\ {\cdot}081 \\ \hline 4{\cdot}345 \end{array}$$

$$\begin{array}{r} 4{\cdot}345 \\ 2{\cdot}896\,3 \\ \hline 1{\cdot}448\,7 \end{array} \qquad \frac{{\cdot}045}{{\cdot}050} \times 1{\cdot}448\,7 = 1{\cdot}303$$

$$\frac{{\cdot}165}{1{\cdot}5} \times {\cdot}648 = {\cdot}071\,3$$

$$\begin{array}{r} 2{\cdot}889\,8 \\ 1{\cdot}303 \\ \hline 4{\cdot}192\,8 \end{array} \qquad \frac{{\cdot}165}{1{\cdot}5} \times {\cdot}736 = {\cdot}081$$

The residuary resistance of the model appears to be 8½ per cent. higher than that of Taylor's standard series at speed-length ratio of ·895.

APPLICATION OF TAYLOR'S CONTOURS FOR RESIDUARY SERIES A, WITH DIMENSIONS MODIFIED FOR COM-

$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	$\frac{V}{\sqrt{L}}$.	Residuary resistance in lbs. per ton Δ corresponding to values of $\frac{B}{H}$.		
		2·250.	3·750.	2·735.
201 Prismatic = ·564	1·150	12·2	16·3	13·525
	1·200	20·6	23·8	21·635
	1·164	..	..	15·795
116 Prismatic = ·566	1·100	7·25	7·8	7·428
	1·050	5·75	6·8	6·09
	1·063	..	..	6·438
87·7 Prismatic = ·571	1·050	5·475	6·2	5·709
	1·000	4·40	5·125	4·634
	1·014	..	..	4·935
73·4 Prismatic = ·569	1·000	4·075	4·477	4·205
	·950	3·147	3·12	3·138
	·984	..	..	3·863
62·1 Prismatic = ·574	1·000	3·960	4·375	4·094
	·950	3·093	3·025 5	3·071 1
	·956	..	..	3·194 0

Resistance per ton Δ to R. E. Froude's Type 4, paring with Taylor's Standard Series.

Working out from Taylor's contours.

$$\begin{array}{r}1{\cdot}200\\1{\cdot}150\\\hline {\cdot}050\end{array}\qquad\begin{array}{r}1{\cdot}164\\1{\cdot}150\\\hline {\cdot}014\end{array}\qquad\begin{array}{r}21{\cdot}635\\13{\cdot}525\\\hline 8{\cdot}110\end{array}$$

$$\frac{{\cdot}014}{{\cdot}050}\times 8{\cdot}11 = 2{\cdot}27\qquad\begin{array}{r}13{\cdot}525\\2{\cdot}27\\\hline 15{\cdot}795\end{array}$$

$$\begin{array}{r}1{\cdot}100\\1{\cdot}050\\\hline {\cdot}050\end{array}\qquad\begin{array}{r}1{\cdot}063\\1{\cdot}050\\\hline {\cdot}013\end{array}\qquad\begin{array}{r}3{\cdot}750\\2{\cdot}250\\\hline 1{\cdot}500\end{array}\qquad\begin{array}{r}2{\cdot}735\\2{\cdot}250\\\hline {\cdot}485\end{array}\qquad\frac{{\cdot}485}{1{\cdot}500}\times{\cdot}55 = {\cdot}178$$

$$\begin{array}{r}7{\cdot}25\\{\cdot}178\\\hline 7{\cdot}428\end{array}\qquad\frac{{\cdot}485}{1{\cdot}50}\times 1{\cdot}05 = 3{\cdot}4\qquad\begin{array}{r}5{\cdot}75\\{\cdot}34\\\hline 6{\cdot}09\end{array}\qquad\frac{{\cdot}018}{{\cdot}050}\times 1{\cdot}338 = {\cdot}348\qquad\begin{array}{r}6{\cdot}09\\{\cdot}348\\\hline 6{\cdot}438\end{array}$$

$$\frac{{\cdot}485}{1{\cdot}500}\times{\cdot}725 = {\cdot}234\qquad\begin{array}{r}6{\cdot}200\\5{\cdot}475\\\hline {\cdot}725\end{array}\qquad\begin{array}{r}5{\cdot}475\\{\cdot}234\\\hline 5{\cdot}709\end{array}$$

$$\frac{{\cdot}485}{1{\cdot}50}\times{\cdot}725 = {\cdot}234\qquad\begin{array}{r}5{\cdot}125\\4{\cdot}40\\\hline {\cdot}725\end{array}\qquad\begin{array}{r}4{\cdot}4\\{\cdot}234\\\hline 4{\cdot}634\end{array}$$

$$\begin{array}{r}1{\cdot}014\\1{\cdot}000\\\hline {\cdot}014\end{array}\qquad\frac{{\cdot}014}{{\cdot}050}\times 1{\cdot}075 = {\cdot}301\qquad\begin{array}{r}4{\cdot}634\\{\cdot}301\\\hline 4{\cdot}935\end{array}$$

$$\begin{array}{r}4{\cdot}477\\4{\cdot}075\\\hline {\cdot}402\end{array}\qquad\frac{{\cdot}485}{1{\cdot}500}\times{\cdot}402 = {\cdot}13\qquad\begin{array}{r}4{\cdot}075\\{\cdot}13\\\hline 4{\cdot}205\end{array}\qquad\begin{array}{r}4{\cdot}205\\3{\cdot}138\\\hline 1{\cdot}067\end{array}$$

$$\begin{array}{r}3{\cdot}147\\3{\cdot}12\\\hline {\cdot}027\end{array}\qquad\frac{{\cdot}485}{1{\cdot}500}\times{\cdot}027 = {\cdot}008\,73\qquad\begin{array}{r}3{\cdot}147\\{\cdot}008\,73\\\hline 3{\cdot}138\,27\end{array}\qquad\begin{array}{r}{\cdot}984\\{\cdot}950\\\hline {\cdot}034\end{array}$$

$$\frac{{\cdot}034}{{\cdot}050}\times 1{\cdot}067 = {\cdot}725\qquad\begin{array}{r}3{\cdot}138\\{\cdot}725\\\hline 3{\cdot}863\end{array}$$

$$\begin{array}{r}4{\cdot}375\\3{\cdot}960\\\hline {\cdot}415\end{array}\qquad\frac{{\cdot}485}{1{\cdot}50}\times{\cdot}415 = {\cdot}134\qquad\begin{array}{r}3{\cdot}960\\{\cdot}134\\\hline 4{\cdot}094\end{array}\qquad\begin{array}{r}4{\cdot}094\\3{\cdot}071\,1\\\hline 1{\cdot}022\,9\end{array}$$

$$\begin{array}{r}3{\cdot}093\\3{\cdot}025\,5\\\hline {\cdot}067\,5\end{array}\qquad\frac{{\cdot}485}{1{\cdot}50}\times{\cdot}067\,5 = {\cdot}021\,9\qquad\begin{array}{r}3{\cdot}093\\{\cdot}021\,9\\\hline 3{\cdot}071\,1\end{array}$$

$$\frac{{\cdot}006}{{\cdot}050}\times 1{\cdot}022\,9 = {\cdot}122\,9\qquad\begin{array}{r}3{\cdot}071\,1\\{\cdot}122\,9\\\hline 3{\cdot}194\,0\end{array}$$

TABLE XXIII.—MR TAYLOR'S OPTIMUM LENGTH OF MIDDLE BODY AND RESIDUARY RESISTANCE CORRESPONDING.

Lbs. residuary resistance per ton Δ.	Percentages of parallel body.															
	35 per cent.		30 per cent.		25 per cent.		20 per cent.		15 per cent.		10 per cent.		5 per cent.		0 per cent.	
	Pris. coef.	$\frac{V}{\sqrt{L}}$.	Pris. coef.	$\frac{V}{\sqrt{L}}$.	Pris. coef.	$\frac{V}{\sqrt{L}}$.	Pris. coef.	$\frac{V}{\sqrt{L}}$.	Pris. coef.	$\frac{V}{\sqrt{L}}$.	Pris. coef.	$\frac{V}{\sqrt{L}}$.	Pris. coef.	$\frac{V}{\sqrt{L}}$.	Pris. coef.	$\frac{V}{\sqrt{L}}$.
·75	...	...	·771	·55	·739	·56	·712 5	·575	·691	·59						
1·0	...	...	·76	·61	·729	·63	·704	·655	·68	·675						
1·5	·65	·801 5	·76	·656	·729	·702	·701 5	·725								
2·0	·68	·803	·764	·71	·731 5	·703 5	·705	·765	·683	·79						
3·0	...	...	·771	·745	·74	·775	·715	·800 6	·694	·826						
4·0	...	...	·777	·77	·748 6	·805	·726	·835	·707	·861	·688 5	·866				
5·0	...	...	·783	·78	·757	·825	·737	·864	·716	·88	·698	·886	·68	·886		
6·0	...	...	·789	·80	·764	·84	·744 5	·88	·723	·895	·703	·900 4	·685	·900 3		
7·0	...	...	·793	·815	·77	·859	·749 5	·89								
8·0	...	...	·798	·82	·777	·87	·755	·900 5								
9·0	...	...	·801	·83	·782 5	·88	·761	·916	·735	·925						
10·0	...	...	·811	·85	·789	·89	·768	·925	·74	·931	·72	·935				
11·0	...	...	...	...	·795	·900 5	·771 5	·935	·745	·943	...	...	·704	·941	·69	·945
12·0	...	...	...	...	·80	·915	·775	·941								
13·0	...	...	...	...	...	...	·779	·946								

The gaps can only be filled by reference to Mr Taylor's curves—they must not be interpolated arithmetically from the above figures.

Given the progressive trial of a coasting steamer 218 × 32·8 × 9·72 mean draft at trial. The steamer was intended for a draft of about 10 ft. fully loaded. The result is poor, because of excessive propeller slip with insufficient immersion.

Revolutions = 104 per min. at full speed. Displacement = 1 370 tons. Block coefficient = ·69. Mid-area coefficient = ·95. Prismatic coefficient = ·727.

	Data.			Derived results.					
							Skin H.P. per		
$\frac{V}{\sqrt{L}}$.	Knots.	I.H.P.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Skin H.P.	E.H.P.	Residuary H.P.	1 000 sq. ft.	Wetted skin.	Residuary resistance, lbs. per ton Δ.
							From Table IX	Calculated.	
·475	7	232	182	60·5	92·9	32·4	7·05	7·1	1·098
·543 5	8	332	190	88·5	133	44·5	10·39	10·4	1·32
·611	9	493	18	123·3	197	73·7	14·4	14·5	1·95
·679	10	720	172	166	288	122	...	19·5	2·9
·685	10·1	765	166	171	306	211·5	...	20·5	3·18

The E.H.P. is taken at ·42 × I.H.P. throughout, a low propulsive efficiency because of the poor immersion and insufficient surface of the screw. The skin H.P. is taken from Table IX, based upon Tables VII and VIII.

$$\text{Skin H.P.} = f \times \text{wetted surface} \times {\cdot}003\,070\,7 \times V^{2{\cdot}83}$$

$$f = {\cdot}009\,40 \text{ for 218-ft. ship.}$$

Wetted surface = 8 510.

$$\text{Residuary resistance in lbs.} = \frac{\text{Residuary H.P.}}{\text{Speed in knots} \times {\cdot}003\,070\,7}.$$

A "similar ship" would be 220 × 33 × 10 ft. 0½ in. load draught. Mean draught at trial 9 ft. 11 in. Displacement = 1 401 tons. Propeller, 4 blades cast iron. D = 9·5. Pitch = 14·25 ft. Experimental area = 41 sq. ft. Maximum I.H.P. on trial 810, at 10·15 knots, 29·9 per cent. apparent slip. 103 revolutions.

Cylinders $\frac{17-28-45}{33}\times 160$ lbs. W.P. Mean pressure referred to L.P. cylinder = 29·7 lbs. per sq. in.

218-ft. coasting steamer reduced to 100-ft. model in salt water. $100\times 15{\cdot}05\times 4{\cdot}46$.

Using Table XIII, $l^3 = 10{\cdot}36$. Displacement $= \frac{1\,370}{10{\cdot}36} = 132{\cdot}2$.

$l^2 = 4{\cdot}752$. Wetted surface $= \frac{8\,510}{4{\cdot}752} = 1\,790$.

Knots.	Skin H.P.	Residuary H.P.	E.H.P.	I.H.P. assuming $\frac{\text{E.H.P.}}{\text{I.H.P.}} = {\cdot}50$.	Residuary resistance, lbs. per ton Δ.
4·75	4·4	3·63	8·03	16·06	1·098
5·435	6·4	5·08	11·48	22·96	1·32
6·11	8·95	8·08	17·03	34·06	1·95
6·79	12·05	12·7	24·75	49·50	2·9
6·85	12·33	13·83	26·16	52·32	3·18

$$f = {\cdot}009\,70.$$

$$\text{Skin H.P.} = {\cdot}009\,70\times 1\,790\times {\cdot}003\,070\,7\times V^{2{\cdot}83}$$

$$= {\cdot}053\,4\times V^{2{\cdot}83}.$$

$$\text{Residuary H.P.} = \frac{\text{Corresponding residuary H.P. of 218-ft. ship}}{l^{3{\cdot}5}}$$

$$l^{3{\cdot}5} = 15{\cdot}28.$$

Let us see how these residuary resistances per ton of displacement agree with the values obtained from Taylor's contours.

We have $\frac{\text{Beam}}{\text{Draft}} = \frac{B}{H} = \frac{32{\cdot}8}{9{\cdot}72} = 3{\cdot}375$. $\frac{\Delta}{\left(\frac{L}{100}\right)} = 132{\cdot}2$. Midarea coefficient = ·95. Prismatic coefficient = ·727. The new $\frac{\text{Beam}}{\text{Draft}}$ ratio $= \frac{B}{H_1} = \frac{B}{H}\times\frac{{\cdot}926}{{\cdot}95} = 3{\cdot}375\times\frac{{\cdot}926}{{\cdot}95} = 3{\cdot}29$.

From Taylor's contours we obtain the following:—

$\frac{V}{\sqrt{L}}$.	Lbs. residuary resistance per ton Δ.	
	$\frac{B}{H} = 2{\cdot}25$.	$\frac{B}{H} = 3{\cdot}75$.
·60	·797	1·214 7
·65	1·075 6	1·763 3
·70	1·531 4	2·188

By interpolation and extrapolation we deduce the following:—

$\frac{V}{\sqrt{L}}$.	$\frac{B}{H} = 2{\cdot}25$.	$\frac{B}{H} = 3{\cdot}75$.	$\frac{B}{H} = 3{\cdot}29$.
·475	·91	·91	·91
·543 5	·945	1·00	·985
·611	·995	1·30	1·20
·679	1·275	2·03	1·81
·685	1·35	2·08	1·86

Our Plate 11 shows 23 per cent. of parallel body to be usual for this vessel at full speed. Taylor's figs. 125 and 126 show 24 per cent. for minimum residuary resistance, and that either 34 per cent. or 13 per cent. parallel body gives residuary resistance 10 per cent. above the minimum. Taylor's fig. 128 shows that the average residuary resistance per ton of displacement for the speed and parallel body referred to is about 1·6 lb.

Our progressive trial results show that the residuary resistance of this vessel (about double Taylor's value) is evidently considerably augmented by something which we may ascribe to wind, appendages, and very poor propeller efficiency.

If the trials were run on a rough day, the loss of speed would be about 10 per cent., and the increase of power perhaps 30 per cent. Although the propeller pitch ratio is high, a better result would probably have been obtained if a still smaller diameter had been given, a higher pitch ratio, and more blade area. On an even keel at the draught stated, the propeller was not wholly immersed.

The Influence of "Form" upon Resistance.

Note Mr W. Froude's dictum from Biles, I.N.A., 1912, "Hantonia."

In Naval-Constructor D. W. Taylor's paper entitled "Some Model Basin Investigations of the Influence of Form of Ships upon their Resistance," read before the American Society of Naval Architects and Marine Engineers in 1911, results were given from the series of models in which the midship section was common throughout, the form of the ends being varied. With each series four different curves of sectional areas were used, and four different water-lines. Each curve of sectional area combined with each water-line resulted in sixteen models for each series. The models were made in halves, so that each bow could be combined with each stern, making in all 256 possible combinations for each series, —not all experimented upon, though a great many were tried in the tank. Mr Taylor's curves show that the effect on the resistance of considerable variations of form of the models in each series is not great with these vessels, the block coefficient being ·563 and ·600. The following tables give some of the results up to a speed-length ratio = 1·12, beyond which it is not likely that vessels of these forms would be driven in actual practice, though Mr Taylor's curves are carried to higher speeds.

Mr Taylor's Series No. 29. 20-ft. models in fresh water. Beam = 2·795 ft. Draught = 1·118 ft. $\frac{36}{35}\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 129{\cdot}15$. $\Delta = 2\,250$ lbs. Mid-area coefficient = ·960. Prismatic coefficient = ·600. Block coefficient = ·563. $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}5$. $\frac{\text{Length}}{\text{Beam}} = 7{\cdot}15$. Beam 13·98 per cent. of length.

In Mr R. E. Froude's notation, M = 6·05, B = ·845, D = ·338.

					Fine-ended sectional area combined with full-ended water-line. Froude's "skin constant" S=6·48. Wetted surface=70·7. Model No. 1107.			Full-ended sectional area combined with fine-ended water-line. Froude's "skin constant" S=6·505. Wetted surface=71. Model No. 1092.		
Speed in knots.	(K)	$\frac{V}{\sqrt{L}}$.	Froude's L.	$L^{-\cdot175}$.	Total resistance in lbs.	(C)	$OSL^{-\cdot175}$.	Total resistance in lbs.	(C)	$OSL^{-\cdot175}$.
2·0	1·161	·448	·473	1·139	2·6	·86	·845	2·85	...	·85
2·5	1·454	·56	·591	1·093	4·1	·863	·812	4·4	...	·816
2·75	1·599	·616	·65	1·079	5·1	·889	·801	5·3	...	·805
3·0	1·744	·672	·71	1·061	6·3	...	·789	6·3	...	·792
3·25	1·89	·728	·769	1·046	7·5	...	·776	7·5	...	·78
3·5	2·032	·784	·827	1·033	8·95	...	·767	8·95	...	·771
3·75	2·18	·84	·886	1·021	10·5	...	·759	10·5	...	·762
4·0	2·328	·895	·945	1·011	12·65	...	·751	12·65	...	·755
4·25	2·47	·952	1·005	·999	16·2	...	·742	15·6	...	·745
4·5	2·62	1·008	1·065	·989	20·05	...	·735	17·7	...	·738 5
4·75	2·76	1·063	1·122	·978 6	22·8	...	·726	19·9	...	·73
5·0	2·908	1·12	1·182	·971	25·7	...	·721	23·7	...	·725

$$(C) = \frac{r}{171{\cdot}7v^2} \times 232{\cdot}5 = 1{\cdot}354\frac{r}{v^2}.$$

The model with these ends, the bow being of the U or bulbous type, is the worst at speeds below $\frac{V}{\sqrt{L}} = {\cdot}895$, and the best for speeds $\frac{V}{\sqrt{L}} = {\cdot}895$ to $1{\cdot}12$.

Mr Taylor's Series No. 32. 20-ft. models in fresh water. Beam = 2·708 ft. Draught = 1·083 ft. $\frac{36}{35} \frac{\Delta}{\left(\frac{L}{100}\right)^3} = 129·15$. Midship-area coefficient = ·960. Δ = 2 250 lbs. Prismatic coefficient = ·640. Block coefficient = ·600. $\frac{\text{Beam}}{\text{Draught}} = 2·5$. $\frac{\text{Length}}{\text{Beam}} = 7·395$. Beam 13·53 per cent. of length.

					Fine-ended sectional area combined with full-ended water-line. Froude's "skin constant" S=6·43. Wetted surface=70·1. Model No. 1206.			Full-ended sectional area combined with fine-ended water-line. Froude's "skin constant" S=6·65. Wetted surface=72·5. Model No. 1191.		
Speed in knots.	(K)	$\frac{V}{\sqrt{L}}$.	Froude's L.	$L^{-·175}$.	Total resistance in lbs.	(C)	$OSL^{-·175}$.	Total resistance in lbs.	(C)	$OSL^{-·175}$.
2·0	1·161	·448	·473	1·139	2·9	...	·839	2·9	...	·869
2·5	1·454	·56	·591	1·093	4·4	...	·806	4·4	...	·834
2·75	1·599	·616	·65	1·079	5·35	...	·795	5·35	...	·823
3·0	1·744	·672	·71	1·061	6·3	...	·783	6·4	...	·81
3·25	1·89	·728	·769	1·046	7·55	...	·771	7·7	...	·798
3·5	2·032	·784	·827	1·033	9·0	...	·762	9·25	...	·789
3·75	2·18	·84	·886	1·021	10·65	...	·753	11·0	...	·779
4·0	2·328	·895	·945	1·011	13·0	...	·745	13·0	...	·772
4·25	2·47	·952	1·005	·999	17·3	...	·736	16·4	...	·761
4·5	2·62	1·008	1·065	.989	23·5	...	·729	20·6	...	·754
4·75	2·76	1·063	1·122	·978 6	27·65	...	·721	23·65	...	·746
5·0	2·908	1·12	1·182	·971	31·2	...	·716	26·8	...	·740 5

$$(C) = \frac{r}{171·7v^2} \times 232·5 = 1·354\frac{r}{v^2}.$$

The model with these ends, the bow being of the U or bulbous type, is the worst at speeds below $\frac{V}{\sqrt{L}} = ·895$, and the best for speed $\frac{V}{\sqrt{L}} = ·895$ to 1·12.

TABLE XXIV.

Block coef.	Desig'd speed $\frac{V}{\sqrt{L}}$	Length of parallel body as percentage of length of ship.	Curve of cross-section areas.	Fore-body water-line.	Nature of forward end, transverse section.
·86 to ·76	·50 to ·60	Minimum resistance with 38%; 3% greater resistance with 52%, to Minimum resistance with 31%; 3% greater resistance with 44%.	Round at ends.	Round. "In other words, easy buttocks at each end rather than full below and fine above" (Prof. Sadler).	Round V'd rather than U'd.
·73 to ·63	·63 to ·85	Minimum resistance with 28%; 3% greater with 40%, to Minimum resistance with 10% parallel body; 3% greater with 18%. About six-tenths of parallel mid body in fore body.	Hollow forward end "with a given set of dimensions and displacement, a long parallel body forward, with a fine bow, but more gradual diminution aft" (Sadler).	Hollow forward end; rounder aft. Hollowness of water-line forward confined to about 15% of ship's length from bow.	U'd or "clubbed."
·63 to ·53	·85 to 1·10	Minimum resistance with 10% parallel body; 3% greater with 18%, diminishing to 0 at ·6 block coefficient, below which parallel mid body seems undesirable.	Slightly hollow forward.	Slightly hollow forward end.	U'd.
·53 to ·45	1·10 to 1·35	No parallel mid body.	Hollow forward and aft.	Straight, especially above $\frac{V}{\sqrt{L}} = 1{\cdot}2$.	V'd.

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Mr Taylor's conclusions are that, on the whole, the curve of sectional areas at the stern may be varied considerably without materially affecting the resistance, and that they also bear out the truth of Mr W. Froude's dictum laid down forty years ago, viz. that, broadly speaking, the U bow and the V stern were favourable for propulsion, the U transverse section being the equivalent of a fine water-line, and the V transverse section, full on the water-line, the equivalent of fine-ended curve of sectional area. Mr Taylor's experiments with these models seem to show that for speed-length ratios $\left(\frac{V}{\sqrt{L}}\right)$ above ·95 the fine water-line aft is best for easy propulsion.

These vessels were intended for a speed not much above that corresponding to 4 knots for the 20-ft. model. Up to that speed, the differences of resistance accompanying radical variations of form were not great. At speeds higher than the critical speed, viz. a little above 4 knots of the 20-ft. model, the results with the different forms change considerably.

Prismatic coefficient should be fine for speeds up to about the square root of the length, as low as ·50 even. As the speed is increased the best prismatic coefficient rises, until, when a speed of twice the square root of the length is reached, a speed which is attained by only special vessels, where the most favourable prismatic coefficient is more in the region of ·64.

Low prismatic coefficient usually means full midship section, with which hollow water-lines seem necessary, and up to speeds in knots equal to the square root of the length, hollow water-lines are better than straight lines.

Fineness appropriate to Speed.

The recent exhaustive investigations by Mr R. E. Froude, Mr G. S. Baker, Naval-Constructor D. W. Taylor, and Professor Sadler, on the effect of variations in the lines, with constant displacement and dimensions, alterations of the ratio $\frac{\text{Length entrance}}{\text{Length run}}$, and other modifications of the longitudinal distribution of displacement affecting the exact sharpness or shape of one or both ends of the ship, clearly show the importance of longitudinal or prismatic coefficient, though the latter must not be taken as varying directly with resistance for given speed-length ratio. In many cases the importance of prismatic coefficient is secondary, though there is distinctly a suitable prismatic coefficient for each speed required. That it is quite peculiar to

the type of vessel under consideration is summarised in *The Engineer*, 24th April and 10th July 1914, referring to Mr Taylor's conclusions: For speed-length ratio up to 1·1 the best longitudinal coefficient is from ·5 to ·55; above this point the coefficient rapidly increases, reaching about ·65 at $\frac{V}{\sqrt{L}} = 1 \cdot 5$, *i.e.* approaching destroyer speeds,—and a little greater at higher speeds. Taking Mr Taylor's 400-ft. ships, in pairs of equal displacement, the only difference between the two of each pair being in the midship area and longitudinal coefficient, "at 21 knots, No. 10 model, with ·64 longitudinal coefficient, had 2·3 times the residuary resistance of its mate, No. 9, which had ·56 longitudinal coefficient; but when the speed was increased to 24½ knots, their residuary resistances were equal. With another pair, No. 4, of ·64 prismatic coefficient, the resistance at 21 knots was nearly twice that of No. 3, having ·56 coefficient; but at 25½ knots they also coincided, while at still higher speed the model with the fuller prismatic coefficient had actually the lesser resistance."

Mr Taylor's explanation is that at low speeds a large proportion of the wave-making is done at the extreme ends of the vessel, hence the great benefit of fine ends; but at high speeds the wave is long, and the whole body of the ship takes part in wave-making, the smaller midship section then giving the least resistance. This form is not entirely favoured by shipowners because of its behaviour under sea conditions, and in any case the residuary resistance is only about 20 per cent. of the total, so that even a large saving is a small percentage of the total. The gains in economy are theoretical, and do not take account of the earning power.

In experiments made by Mr W. Froude and others to ascertain the effect, on the total resistance, of adding middle body, models were used representing a series of ships of identical cross-section and identical form of ends. The only difference consisted in the length of parallel body, of uniform transverse section, inserted amidships. Mr W. Froude's ships* were 38·4 ft. beam; 14·4 ft. draught; length of fore body 80 ft.; length of after body 80 ft.; parallel mid body 0 to 340 ft.; total length 160 ft. to 500 ft. Up to 60 ft. middle body the amount was decreased by 10 ft. for each experiment, and over 60 ft. it was decreased successively by 20 ft., by cutting them amidships and rejoining the ends. Curves of resistance in tons were plotted to a base of speed in knots. Adding middle body increased displacement, and at low speeds

* *Transactions Inst. Naval Architects*, 1877.

increased the resistance by the same amount, but as the speed was increased the shorter ships showed greater resistance in many cases. Thus at 14 knots the 280-ft. ship showed less resistance than either the 200-ft. ship or the 240-ft. ship. At $14\frac{1}{2}$ knots the 360-ft. ship showed almost no more resistance than the 200-ft. ship of 2 275 tons less displacement. Mr R. E. Froude afterwards pointed out that the formation of the stern waves was to some extent arrested by the residue of the bow waves, and this was the cause of the humps and hollows.

CRITICAL SPEEDS.

The speed at which the residuary resistance first rapidly begins to increase depends principally upon the wave-making features of the vessel. For general purposes it may be considered as proportional to the speed of a wave of length equal to that of the ship. Taking V in knots and length L in feet, we have

$$V \sim \sqrt{\frac{gL}{2\pi}} = C\sqrt{L},$$

where C is a constant.

M. Normand's formulæ for maximum normal speed are as follows:—

$$V = \frac{(1{\cdot}01m - b)L}{1{\cdot}01\sqrt{BH} \times m^{\frac{3}{2}}}$$

or

$$V = \frac{1{\cdot}39(1{\cdot}01m - b)L}{\sqrt[4]{BH} \times m^{\frac{3}{4}}}$$

where B = beam of ship in feet,
H = draught in feet,
L = length in feet,
b = block coefficient,
m = midship section coefficient.

At or about this speed the $\frac{\text{Expanded area}}{\text{Disc area}}$ of the propeller or propellers is given by M. Normand as

$$r^{\frac{3}{4}} = \frac{J \times I.H.P.}{nD^2V^2}$$

where r is the area ratio,
J a constant = 6 to 8,
n = number of propellers,
D = diameter in feet,
V = speed in knots.

As immersion is greater and conditions favourable, J is less. As immersion is less and conditions unfavourable, J is greater.

The following particulars are taken from Mr R. E. Froude's 1898 paper to the I.N.A., on the effect of direction of turning in twin-screws.

TABLE XXV.

Ship.	$\frac{\text{Length}}{\text{Breadth}}$.	Type.	Dimensions of 100-ft. model.	Prismatic coefficient. Fore body.	Prismatic coefficient. After body.	Approx. speed $\frac{V}{\sqrt{L}}$.
1	5·2	Battleships	100 × 19·25 × 7·06	·638	·737	·92
2	5·26		100 × 19 × 6·67	·600	·684	·92
3	5·26		100 × 19 × 6·66	·618	·678	
4	5·63	Cruisers	100 × 17·8 × 6·56	·577	·587	1·03
5	5·63		100 × 17·8 × 6·56	·561	·573	1·03
6	5·63		100 × 17·8 × 6·56	·561	·573	1·03
7	7·15		100 × 14 × 5·27	·568	·628	1·10
8	7·8		100 × 12·83 × 4·63	·640	·704	·80
9	7·69		100 × 13·02 × 4·63	·613	·683	·90
			Lines approaching Atlantic liner type.			
10	7·69		100 × 13·02 × 4·63	·593	·668	·90
11	6·32		100 × 15·83 × 5·79	500	·593	·95
12	6·28		100 × 15·94 × 5·65	·567	·622	
13	8·33		100 × 12·02 × 4·5	·533	·620	1·05
14	8·24		100 × 12·18 × 4·5	·569	·596	1·03
15	7·63		100 × 13·11 × 5·5	·531	·607	1·00
			Thornycroft pattern of stern.			
16	5·55	Extreme light draft	100 × 18·07 × 4·16	·577	·624	
17	10·8	Destroyers (Thornycroft)	100 × 9·26 × 2·72	·573	·540	1·92
18	10·72	,,	100 × 9 33 × 2·84	·531	·581	1·91
19	9·99	(Laird)	100 × 10·13 × 2·78	·535	·605	
20	10·0		100 × 10·0 × 2·65	·505	·544	1·93
21	10·5	(Palmer)	100 × 9·54 × 2·69	·595	·613	2·00

The humps and hollows on the resistance curves of similar ships occur at similar speeds. In a general way it has been recognised since about 1880 that the deeper the draught the higher are the speeds at which the humps and hollows appear. Mr R. E. Froude, in his 1881 paper, gave the hump speeds and the hollow speeds for a series of ships. Taking them as 100-ft. models, we have—

Hump speeds, 6·05, 6·85, 8·1, 10·45, and 18 knots.
Hollow speeds, 6·4, 7·4, 9·05, and 12·8 knots.

In the resistance or horse-power curves of very fine vessels the humps and hollows are not so pronounced as in those of fuller vessels. On the other hand, long parallel body and fine ends are frequently associated with humpy resistance curves.

As the skin frictional part of the resistance varies uniformly according to the expression $f . S . V^n$, it is only the residuary resistance curve that is humpy. The humpiness of the I.H.P. or E.H.P. curve partakes of the sinuous character of the curve of residuary resistance, after the latter has been separated from the skin frictional element of the total resistance. So far as the I.H.P. curve is concerned, the appropriate limit of speed, or "limiting economical speed," has frequently been considered to be the speed at which the I.H.P. is varying as about the fourth power of the speed. This point may be found by trial, by drawing tangents to the speed-power curve, or by the logarithmic method given on p. 88. At higher speeds the I.H.P. may vary as the seventh or eighth or a still higher power of the speed. In our progressive trials the limiting economical speed is in some cases marked by an arrow, a survival from our first edition, in which an attempt was made to name the limiting economical speeds in nearly all cases. In a paper read before the Institution of Naval Architects in 1901, Sir E. Tennyson D'Eyncourt pointed out that, usually about 12 per cent. above the limiting economical speed, the I.H.P. varied as the seventh power of the speed, whilst the wave horse-power varied as V^7 at the limiting speed, and as V^{10}, or sometimes as a higher power of V, at about 12 per cent. above the limiting economical speed, giving percentage ratios of skin horse-power and wave horse-power for these speeds for average vessels of considerable beam but having fine entrance and run and full midship section.

Colonel Rota, in his researches at the Italian experimental tank, showed some effects of modifying one dimension at a time, length, and breadth and draught successively, keeping one speed. Increase of length, up to a certain point, was shown to reduce the wave-making, though it increased the skin friction. Various

displacements secured equal speeds with equal powers. After developing the dimensions up to 6 000 tons for a ship of 18 to 22 knots speed, sometimes a reduction of length, though it reduced the displacement, required an increase of power for a given speed. Some of the effects upon wave-making resistance of variations in the longitudinal distribution of displacement have been ably expounded by Professor Sadler and Mr D. W. Taylor, and are dealt with on pp. 111–114. The shipowner usually requires a vessel of a certain length, displacement, and speed to fulfil the conditions of the service, and, according to Mr R. E. Froude's dictum, the resistance at that speed is almost solely determined by the shape of the curve of cross-section areas, including prismatic coefficient, by the extreme beam, and by the water-line, particularly of the fore body. Mr Taylor and Professor Sadler have shown how varying the shape of the lines of the entrance and run, and giving various percentages of parallel body, affect the residuary resistances for appropriate speeds. Mr G. S. Baker, at the National Physical Laboratory experiment tank, has shown how wave-making effects vary directly as the length and prismatic coefficient by a new law indicating the manner in which resistance due to transverse wave-making varies with length, speed, and prismatic coefficient, and has deduced a method of determining economic length of parallel body (p. 124).

ANGLES OF ENTRANCE AND RUN.

When making comparisons with a view to determining the necessary "sharpness" of the ends of a ship, it is convenient to have a formula for the angles of entrance and run. M. Normand stated that the first factor, viz. $\frac{0 \cdot 96SL - W}{8^{\frac{3}{2}}}$ of his formula referred to later, was inversely proportional to the tangent of the angles of the longitudinal stream lines.* A list of the values of this first factor is given below for a few known ships. (See other table for their dimensions.)

Another formula, based, however, on Kirk's analysis only, is the following, given by Mr W. Hök in the discussion on his valuable paper to the North-East Coast Institution of Engineers and Shipbuilders in 1893.

$$\tan \theta = \frac{\psi}{1 - \phi} \times \frac{B}{2L}.$$

* There is scope for investigation of this relation. Want of space prevents our attempting it here, but any reader would find it worth his while to make the necessary comparison with the lines.

Where ψ = coefficient of midship section; ϕ = prismatic coefficient of displacement; B = breadth of ship; L = length of ship; and $\theta \times \frac{1}{2}$ mean angle of entrance and run.

This is a useful formula, but it must be remembered that the angle found is not the angle of the ends of the ship at the water-line, but the angle for Kirk's block model, approximately the *mean* angle.

In the following list the results of the two formulæ are placed side by side:—

Ship.	Hök's formula based on Kirk's model.		Normand's first factor.
	tan θ.	Degrees.	
Normannia	·144 3	8·2	13·75
Iris	·151 8	8·633	10·43
M——	·152 1	8·65	13·17
S.S. passenger steamer .	·157	8·92	11·83
Hammonia . . .	·162 7	9·25	12·1
Yorktown . . .	·166	9·433	10·4
Chicago . . .	·182	10·317	10·51
Ceram	·190 5	10·784	10·7
Cincinnati . . .	·195	11·033	11·44
184-ton yacht . .	·201	11·38	9·45
Lepanto . . .	·239 3	13·467	8·35
T.S.S. 1906 . . .	·255	14·3	12·5
P——	·262	14·685	10·75
Bayern . . .	·363	19·95	9·36

The first formula is not so useful as the second when drawing the lines, but it shows at once which boats are easy to drive.

Normand's Normal Speed.

In a paper read before the Institution of Naval Architects in 1888, "On the Fineness of Vessels in Relation to Size and Speed," M. Normand defined the "normal speed" as the speed which can be obtained without any undue waste of power, and said that this speed increases for a given size with the fineness of the longitudinal stream lines. It also increases with the size for a given fineness.

The following is the formula given by Normand in 1870, quoted again in his 1888 paper, and proved :—

U = normal maximum speed ;
L = length of vessel on load line ;
S = area of immersed section ;
W = displacement ;

$$U = a \times \frac{0 \cdot 96SL - W}{S^{\frac{3}{2}}} \times S^{\frac{1}{4}}.$$

M. Normand states that the second factor, $S^{\frac{1}{4}}$, is proportional to the square root of the linear dimensions of the immersed midship section ; and he remarks that, in applying the formula to different types of vessels, it will be seen that the coefficient a increases with the draught, *e.g.* a "normal speed" greater than that indicated by the formula may be had from an ironclad of 28 ft. draught.

It may be instructive to compare this "normal speed" with our "limiting economical speed" by finding the value of the coefficient a for some examples from our list of vessels tried progressively.

To simplify the arithmetic, we shall consider only the 100-ft. model in each case.

$S^{\frac{3}{2}}$ means the square root of the cube of S, or the cube of the square root of S ; and

$S^{\frac{1}{4}}$ means the fourth root of S, or the square root of the square root of S.

Name.	S.	$S^{\frac{3}{2}}$.	$S^{\frac{1}{4}}$.	Name.	S.	$S^{\frac{3}{2}}$.	$S^{\frac{1}{4}}$.
Normannia .	46·9	321·2	2·62	Chicago .	80	715·5	2·99
M—— . .	51	364·2	2·672 2	Iris . .	82	742·6	3·01
Hammonia .	60·5	470·9	2·79	Yorktown .	82·6	750·6	3·015
Passenger S.S.	63·3	503·6	2·82	184-ton yacht	99·4	991·1	3·16
Cincinnati .	67·6	555·8	2·87	Bayern .	100·7	1 010·3	3·17
P—— . .	76·4	667·9	2·96	Lepanto .	127	1 433	3·355
Ceram . .	77·6	683·6	2·97				

APPLICATION OF M. NORMAND'S FORMULA FOR THE NORMAL MAXIMUM SPEED OF SHIPS.

Reduced Typical Vessels.

Name.	Tons displacement.	Length.	Beam.	Mean Draft.	Block coefficient.	Midship area coefficient.	Knots.	Prismatic coefficient.	Speed at which I.H.P. varies as V^4.	Value of a (in Normand's formula) which gives same speed.
T.S.S. 1906	122·1	100	12·69	4·72	·716	·932	7·5	·768	6·8	·199
Cincinnati	133·2	100	19·6	5·6	·425	·615 4	7·38	·69	6·95	·212
P——	158	100	15·1	5·46	·67	·926	8·0	·734	7·0	·22
M——	98·1	100	11·64	5·17	·573	·846	7·33	·676	7·25	·206
Bayern	223	100	18·7	6·11	·682	·882	7·99	·773	7·5	·253
Hammonia III.	114	100	12·06	5·42	·609	·927	7·9	·657	7·56	·224
Single-screw passenger steamer	111	100	12·29	5·49	·594	·938	8·99	·633	8·25	·247
Chicago	145·6	100	15·3	6·03	·551	·868	8·65	·635	8·25	·262
Normannia	85·5	100	11·5	4·46	·582	·915	9·32	·636	8·5	·236
184-ton T.S. yacht	184	100	21	7·06	·435	·67	10·29	·65	9·0	·301
Lepanto	230	100	18·17	7·53	·59	·896	9·5	·66	9·0	·321
Ceram	145·2	100	16·85	5·89	·513	·783	9·89	·654	9·7	·305
Iris	123	100	15·4	6·0	·488	·889	10·72	·549	10·25	·327
Yorktown	138·3	100	15·67	6·09	·513	·867	11·0	·591	10·75	·343

Note.—As nearly all steamers are run at a speed a small percentage in excess of their "limiting economical speed" (the "limiting economical speed" being understood to mean the speed at which I.H.P. varies as V^4), another column of values of a may of course be obtained and used for estimating this higher speed by means of M. Normand's formula. It is interesting to note that the same values of a are obtained with the actual ship dimensions as with the 100-ft. model dimensions, the latter being preferable, as it entails less arithmetic.

Absolute size in relation to speed.—Mr Hillhouse's table of $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$ for trial trip conditions for given L.W.L., and our Plate 39 showing appropriate values of $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$ for ships of various lengths on voyage, are for ships whose length is favourable for the intended speed. For the fulfilment of the owners' requirements regarding speed and carrying capacity, tank experiments may point to one size of ship and the owners' experience to another. The length, beam, and draught of ship are usually prescribed by the owners, who require a certain carrying capacity, which means a certain coefficient of fineness, and the speed for the particular trade may or may not be the optimum for that prismatic coefficient from the point of view of the experimenter. When a tank expert is consulted he is not always given the opportunity of deciding the principal dimensions of the proposed new ship. In many cases he is only asked to experiment with one or two models to adjust the prismatic coefficient and the longitudinal distribution of displacement to a limiting speed which will lead to efficient propulsion. Mr Baker has, in his papers,* shown how, by the use of the (P) value, the relation between length, speed, and prismatic coefficient can be arrived at. The (P) values correspond to hollows in the resistance curve. When the length, speed, displacement, and midship section are fixed, the experimenter can do little except try different lengths of entrance and run and parallel body. If, after the ship is built and put in commission, $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$ turns out to be 230 when it might have been 280, if the coal per I.H.P. hour is as low as possible, it might be that a different size of ship could be propelled more economically at the given speed. Methodical experiments in rough water and wind can show the direction modifications should take, and possibly speeds could be decided upon suitable for the proportions, while an adjustment of suitable coefficients of fineness and corresponding limiting speed would produce greater efficiency. By the use of Mr Baker's (P) values, the theoretical limiting speed can be estimated.

The following is an example of the use of the "constant"

* *Transactions Inst. Naval Architects*, 1913, G. S. Baker on "Methodical Experiments with Mercantile Ship Forms"; and 1915, J. L. Kent on "Further Model Experiments on the Resistance of Mercantile Ship Forms: The Influence of Length and Prismatic Coefficient upon the Resistance of Ships."

system of notation used by Mr R. E. Froude and Mr G. S. Baker.

Example.—Let us suppose that we are beginning to design a single-screw cargo steamer 340 × 46·5 ft. × 23 ft. 4 in. mean draught fully loaded. Block coefficient = ·76. 8 000 tons displacement. $\dfrac{\Delta}{\left(\dfrac{L}{100}\right)^3} = 203{\cdot}8.$

If midship section coefficient = ·975, then $\dfrac{\cdot 76}{\cdot 975} = \cdot 78$ = prismatic coefficient.

The vessel is required to maintain a speed of not less than 10 knots when fully loaded to 23·33 ft. draught, and 10½ knots when partly loaded, say, to 19 ft. 6 in. mean draught, about 6 670 tons displacement. First let us see if these are theoretical speeds for the form proposed. Hump speeds should be avoided, and speeds should be chosen which lie rather in the hollows of the resistance curve, speeds at which a rise in the resistance is just beginning. From Mr G. S. Baker's papers we find that the critical speed of any ship is given by the expression

$$V = 1{\cdot}34\sqrt{\frac{P \times L}{n}}$$

where n is the number of wave crests between the bow and stern systems of waves, P = prismatic coefficient, L = length of ship in feet, and V = speed of ship in knots.

$$V = 1{\cdot}34\sqrt{\frac{\cdot 78 \times 340}{4}} = 10{\cdot}9.$$

$$V = 1{\cdot}34\sqrt{\frac{\cdot 78 \times 340}{5}} = 9{\cdot}75.$$

10·9 knots and 9·75 knots are two economical speeds for this vessel.

The propulsive coefficient would in all probability be ·47 at least against calm air only. ·44 is commoner with direct turbines, where the propeller efficiency is low, but in a single-screw cargo steamer with reciprocating steam engines we might take ·47.

A vessel of finer block would perhaps be less adversely affected by rough water, and might maintain greater regularity of service,

but for many trades the difference in carrying capacity with ·76 compared with ·75 is deemed to outweigh this disadvantage.

With ·75 the critical speeds on a smooth-water basis would, of course, be different from those given for ·76. Theoretically they would be lower until a different value of n, the number of wave crests intervening between the bow and stern wave systems, caused V to take a sudden jump, in accordance with Mr G. S. Baker's expression

$$V = 1{\cdot}34\sqrt{\frac{P \times L}{n}}.$$

For every ship the values of the constant denoted by the symbol (P) may be found, denoting the positions of the humps and hollows on the resistance curve.

$$(P) = \frac{V}{\sqrt{P \times L}} \times {\cdot}746.$$

For our vessel, at 10·9 knots,

$$(P) = {\cdot}746 \times \frac{10{\cdot}9}{\sqrt{{\cdot}78 \times 340}} = {\cdot}50,$$

and at 9·75 knots,

$$(P) = {\cdot}746 \times \frac{9{\cdot}75}{\sqrt{{\cdot}78 \times 340}} = {\cdot}447.$$

As our ship has a form somewhere between Mr Baker's (1913) ship D and ship E, we may use his (C) curves plotted upon a base of (P) (Plate 38). The form is a little nearer D than E, say $\frac{2}{5}$ths from D and $\frac{3}{5}$ths from E, roughly, and the parallel body about $35\frac{1}{2}$ per cent. of the ship's length. The range of the ratio

$$\frac{\text{Length of entrance}}{\text{Length of run}} = {\cdot}6 \text{ to } 1{\cdot}66.$$

	Coefficients.		
	Block.	Prismatic.	Mid area.
Set D . .	·739 5	·755	·98
Set E . .	·805	·82	·98
Our ship . .	·76	·78	·975

$\frac{V}{\sqrt{L}}$.	Economical speeds in knots.	(P)	(C) corrected for 340-ft. ship.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$. $\rho = \cdot44$.	$\rho = \cdot46$.	$\rho = \cdot48$.	$\rho = \cdot50$.	(Taking propulsive coefficient = ·44.) Values of I.H.P.
·529	9·75	·447	·825 5	228	238	248	258	1 630 @ 23′ 4″ draught. 1 440 @ 19′ 6″ draught.
·591	10·9	·50	·840 5	223	234	244	254	2 340 @ 23′ 4″ draught. 2 050 @ 23′ 4″ draught.

Where ρ = propulsive coefficient $= \frac{\text{E.H.P.}}{\text{I.H.P.}}$.

Since $(C) = \frac{\text{E.H.P.}}{\Delta^{\frac{2}{3}}V^3} \times 427{\cdot}1$.

If $\rho = \cdot50$, $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \frac{213{\cdot}5}{(C)}$. If $\rho = \cdot44$, $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \frac{188}{(C)}$.

If $\rho = \cdot46$, $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \frac{196{\cdot}5}{(C)}$. If $\rho = \cdot48$, $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \frac{205}{(C)}$.

If I.H.P. = 1 900, we may take engines $\frac{24\text{ in.}-40\text{ in.}-67\text{ in.}}{45\text{ in.}}$ × 180 lbs. W.P. 70 revolutions per min. E_{pm} = 33 lbs. per square inch.

(E_{pm} is a convenient symbol for equivalent mean pressure in lbs. per square inch referred to the L.P. cylinder, used in Seaton and Rounthwaite's *Pocket-Book of Marine Engineering Rules and Tables.*)

With natural draught boilers, three S.E.B. each with three corrugated furnaces give 187·6 sq. ft. of grate, which will provide 1 900 I.H.P. at sea comfortably. With Howden's F.D., 130 sq. ft. of grate would serve the same power equally well. These allowances give a margin for maintaining regular speed when cleaning fires.

As stated above, the economical speeds are 9·75 knots and 10·9 knots. The speeds required, however, are 10 knots and 10½

knots. The difference between 10·9 and 10·5 may be called an allowance for wind, while the lower speed 9·75 is not required.

If Mr G. S. Baker's formula for P may be used for these speeds, we have

$$P = \cdot 746 \times \frac{10\cdot 5}{\sqrt{\cdot 78 \times 340}} = \cdot 481,$$

and

$$P = \cdot 746 \times \frac{10}{\sqrt{\cdot 78 \times 340}} = \cdot 459.$$

Using Mr G. S. Baker's C curves for his ships D and E,
we find C = ·794 for a 400-ft. ship at the speed for P = ·481,
and C = ·816 5 for a 400-ft. ship at the speed for P = ·459.

The correction for C in passing from a ship of 400 ft. long to one of 340 ft. in length is ·010 5, to be added to the above C values.

Therefore we have, for $10\frac{1}{2}$ knots, C = ·804 5 and for 10 knots, C = ·827 0, approximately.

Now if we take the propulsive coefficient as ·44 as before, let us convert the C "constant" into the more flexible formula $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.

$$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \frac{188}{C}.$$

We find

$$\frac{354 \times (10\tfrac{1}{2})^3}{1\,750} = \frac{188}{\cdot 804\,5} = 234,$$

i.e. 1 750 I.H.P. for $10\frac{1}{2}$ knots at 19 ft. 6 in. mean draught, and

$$\frac{400 \times (10)^3}{1\,760} = \frac{188}{\cdot 827\,0} = 227,$$

i.e. 1 760 I.H.P. for 10 knots at 23 ft. 4 in. mean draught.

These C values apply to a clean painted ship running in smooth salt water under good conditions.

Critics may remark that the value of ρ which we have selected, viz. ·44, is on the low side, and that ·46 or ·48 might be expected in a smooth sea. ·445 was the actual value of the ratio $\frac{\text{E.H.P. (naked)}}{\text{I.H.P.}}$ in the case of a 418-ft. twin-screw steamer at

8 000 tons displacement, of the same fleet, and in the absence of tank trial data for the 340-ft. single-screw cargo boat, ·44 is taken as at least a safe propulsive coefficient for average service conditions.

It is necessary, moreover, to provide a margin of power for wind resistance. The upper works of the vessel—the masts, funnel, bridge, wheel-house, deck-houses, and hull exposed to the wind—present a thwartship area of about 1 891 sq. ft. at full load draught, and 2 054 sq. ft. at 19 ft. 6 in. draught.

The amount of the air resistance can be approximately estimated for a given route. Suppose that on the outward run, when a passenger liner is steaming at 14 knots, the smoke rises vertically from the funnel with a following wind, the rate of the wind is about 14 knots, *i.e.* the same speed as the ship; and if that is only a light wind compared with the usual wind on the route—about a 25-knot breeze,—let us estimate the resistance of our 10-knot cargo ship coming up against the trade winds, *i.e.* against a head wind of that speed. Then $V = 10 + 25 = 35$.

$$\begin{aligned} R &= \cdot 004\,3 \times A \times V^2 \\ &= \cdot 004\,3 \times 1\,891 \times 1\,225 \\ &= 9\,950 \text{ lbs.} \\ \text{Air H.P.} &= \cdot 003\,070\,7 \times 9\,950 \times 10 \\ &= 305. \end{aligned}$$

In calm air (no wind),

$$\begin{aligned} V &= 10. \\ R &= \cdot 004\,3 \times A \times (10)^2 \\ &= 814 \text{ lbs.} \end{aligned}$$

The air H.P. required against the 25-knot wind would be about 305, and the horse-power required to overcome calm air resistance would be about 25.

Adding the air horse-power to the I.H.P. already estimated for the hull in smooth salt water, we have 1 781·5 I.H.P. for $10\frac{1}{2}$ knots in calm air at 19 ft. 6 in. draught, $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = 230$, and 1 785 I.H.P. for 10 knots in calm air at 23 ft. 4 in. draught, $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = 224$.

At 19 ft. 6 in. draught $A = 2\,054$ sq. ft. $V = 10\frac{1}{2}$ knots.

$$\begin{aligned} R &= \cdot 004\,3 \times A \times V^2 \\ &= \cdot 004\,3 \times 2\,054 \times (10\tfrac{1}{2})^2 \text{ in calm air} \\ &= 975 \text{ lbs.} \\ \text{Air H.P.} &= \cdot 003\,070\,7 \times 975 \times 10\cdot 5 \\ &= 31\cdot 5. \end{aligned}$$

Against a 25-knot wind $V = 35{\cdot}5$.

$$\begin{aligned} R &= {\cdot}004\,3 \times 2\,054 \times 1\,260 \\ &= 11\,120 \text{ lbs.} \\ \text{Air H.P.} &= {\cdot}003\,070\,7 \times 11\,120 \times 10{\cdot}5 \\ &= 359. \end{aligned}$$

The difference between the air H.P. in calm air and the air H.P. against the 25-knot wind at 10½ knots is $359 - 31{\cdot}5 = 327{\cdot}5$. For the 10-knot condition the difference is $305 - 25 = 280$.

$$1\,781{\cdot}5 - 327{\cdot}5 = 1\,454.$$
$$1\,785 - 280 = 1\,505.$$

$$V^3 = \frac{C \times \text{I.H.P.}}{\Delta^{\frac{2}{3}}}.$$

Against the wind, at 19 ft. 6 in. draught,

$$V^3 = \frac{230 \times 1\,454}{354} = 948.$$

$$\therefore \quad V = 9{\cdot}83 \text{ knots.}$$

Against the wind, at 23 ft. 4 in. draught,

$$V^3 = \frac{224 \times 1\,505}{400} = 845.$$

$$\therefore \quad V = 9{\cdot}46 \text{ knots.}$$

Both with the I.H.P.'s (1 781·5 and 1 785) named above.

In order to maintain the required speeds, however, we must add the difference of air H.P., thus :—

$1\,781{\cdot}5 + 327{\cdot}5 = 2\,109$ I.H.P. for 10½ knots at 19 ft. 6 in. draught,
$1\,785 + 280 = 2\,065$ I.H.P. for 10 knots at 23 ft. 4 in. draught.

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = \frac{354 \times (10{\cdot}5)^3}{2\,109} = 194.$$

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = \frac{400 \times (10)^3}{2\,065} = 194.$$

If the boiler power is only good for 1 950 I.H.P. continuously, the speeds against a 25-knot wind would be 10·23 and 9·73 knots, but the speeds of 10½ and 10 knots could be maintained against a 16½-knot wind, in which the results would be :—

For 10½ knots, air H.P. = 168·5 difference, and
for 10 knots, air H.P. = 165 difference.

In the $10\frac{1}{2}$-knot condition total air H.P. = 200, V = $\sqrt{703}$, R = 6 210 lbs.

In the 10-knot condition total air H.P. = 190, V = $\sqrt{723}$, R = 6 200 lbs.

answering to the usual description of a "fresh wind on the bow."

Therefore the final figures for this condition would be

$$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = 210 \text{ at } 10\tfrac{1}{2} \text{ knots, on 19 ft. 6 in. draught, and}$$

$$= 205 \text{ at 10 knots, fully loaded, on 23 ft. 4 in. draught.}$$

About 20 per cent. of this cargo steamer's I.H.P. is expended in overcoming wind resistance when steaming against a 25-knot wind at full speed at 19 ft. 6 in. draught. At 23 ft. 4 in. draught the percentage is about 17·2. About 11·2 per cent. of the I.H.P. is expended in overcoming wind resistance when steaming at full power against a $16\frac{1}{2}$-knot wind at 19 ft. 6 in. draught. At 23 ft. 4 in. draught the percentage is about 10·7. When there is no wind, the air resistance absorbs about $1\frac{3}{4}$ per cent. of the I.H.P. at full speed. The reduction of speed against a $16\frac{1}{2}$-knot wind would be nearly $\frac{3}{4}$ knot, and against a 25-knot wind would be a knot.

For higher values of the propulsive coefficient than ·44 the results would be correspondingly better.

The propulsive coefficient which we have chosen (·44) is not an uncommon figure with direct turbines where the propeller efficiency is low, but for our single-screw merchant steamer with reciprocating steam engines ·47 could safely be assumed. The figures for I.H.P., etc., would therefore all improve.

Waves causing pitching would naturally increase the resistance at a given speed. The effect of the waves which would be produced by such a wind as the above-mentioned would be considerable. The wind might be accompanied by a head sea, which would be a serious obstacle to the speed of a boat 340 ft. in length, though it would not interfere with the time-keeping of a Transatlantic liner of the largest size.* In a heavy sea, according to *The Engineer*, 4th February 1916, "with a following wind and the same power developed there is an increase of speed over smooth-water conditions so long as the speed of wind does

* "The 'Mauretania' averaged for a whole year, on thirty consecutive passages westward and eastward, in all weathers and under varying and uncontrollable conditions of service, a mean speed of 25·5 knots. Between February and August 1911 the total number of revolutions of the screws during each passage varied only 2 per cent. above or below the number of revolutions per passage deduced from an average for all the passages." (Sir Wm. H. White.)

not exceed 25 knots. At that speed the accompanying waves proper to such a wind increase the resistance sufficiently to balance the advantage gained from the wind pressure, and the speed is the same as for smooth water. With a further increase of speed of wind there is actually a decreased speed of ship."

The humps in the resistance curves of ships of 300 to 500 ft. in length, running at 11 to 15 knots, are those which concern the majority of shipowners. At the lower speeds there is inevitable wave-making resistance due to the diverging waves set up by the bow and the stern, accompanied by minor humps. At the higher speeds, transverse waves are found; and when we reach a certain critical speed, depending upon the shape of the vessel, the resistance curve begins to rise abruptly. Mr G. S. Baker has shown how the lengths of entrance and run should be modified in order that this abnormal rise in resistance may be minimised. He has indicated by approximate formulæ the critical speed and the limiting economical speed.

The critical speed of any ship is given by the expression

$$V = 1{\cdot}34\sqrt{\frac{P \times L}{n}},$$

representing speeds at which there are hollows in the resistance curve, where n is the number of wave crests between the bow and stern system of transverse waves. When $n = 1$, there is one wave crest amidships between the bow and stern systems. At a lower speed, when $n = 2$, 3, or 4, there are two, three, or four wave crests between the first crest of the bow system of waves and the first crest of the stern-wave system.

Using values of V from the formula for critical speeds,

$$\textcircled{P} = {\cdot}746\sqrt{\frac{V}{P \times L}}.$$

L = length of ship in feet.
V = speed in knots.
P = prismatic coefficient.

Mr G. S. Baker's Ⓒ values from tank trials are usually plotted upon a base of Ⓟ.

For the published figures for "Ulysses" and "Achilles"

$$V = 1{\cdot}34\sqrt{\frac{{\cdot}736 \times 514}{3}} = 15{\cdot}03,$$

corresponding to

$$(P) = \frac{1}{\sqrt{3}} = \cdot 577, \quad \text{and} \quad \frac{V}{\sqrt{PL}} = \cdot 774.$$

Again,

$$V = 1 \cdot 34 \sqrt{\frac{\cdot 736 \times 514}{4}} = 13 \cdot 02,$$

corresponding to

$$(P) = \frac{1}{\sqrt{4}} = \cdot 50, \quad \text{and} \quad \frac{V}{\sqrt{PL}} = \cdot 67,$$

since

$$\frac{(P)}{\cdot 746} = \frac{V}{\sqrt{PL}} \text{ for any ship.}$$

"Ulysses" and "Achilles" at 14 knots have (P) = ·538.

For (P) = ·538 in the abscissa, we find the (C) value for the contract speed. A tank trial will show whether this spot lies in a hollow or not. Thus there is not only a certain fineness appropriate to a certain speed, but there is size also to be taken into account, absolute length of vessel together with fineness, as in the term $\sqrt{P \times L}$, where P = prismatic coefficient, and L = length of ship in feet. The speed may be estimated and predicted with some reliability for smooth-water conditions, but whether the fineness and length appropriate to a given speed in smooth water are the best for everyday voyaging in the rough ocean or not is a question which must not be overlooked. An article in *The Engineer*, 4th February 1916, deals with this question of speed of cargo steamers and of sea kindliness. "The experience of ship captains has recently led to the adoption of larger ships with finer lines than formerly, though they are more expensive to construct than shorter, fuller vessels having the same cargo-carrying capacity. . . . Not only is it found that the finer entrance of the larger vessel produces better timekeeping in rough weather than is possible with fuller ships, but it is also the common experience that larger ships keep better time than smaller vessels of similar fineness. Taylor has laid it down that 'the increase of resistance in rough water is, under practical conditions, largely a question of absolute size; waves 150 ft. long and 10 ft. high would not seriously slow a 40 000-ton vessel 800 ft. long. A vessel of 120 ft. long would find them a very serious obstacle to speed.'"

Let us build up the power for the 340-ft. cargo vessel, using Real-Admiral Taylor's curves for residuary resistance in lbs. per ton of displacement.

First.—We have $\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 203{\cdot}8$. Prismatic coefficient = ·78.

At least 10 knots for the fully loaded condition, viz. 23·33 ft. mean draught.

Second.—For the partly loaded condition, 19 ft. 6 in. draught, we require $10\frac{1}{2}$ knots. As this may be more difficult to realise, with a given power, than 10 knots fully loaded, let us take the second case. At $10\frac{1}{2}$ knots, $\frac{V}{\sqrt{L}} = {\cdot}57$. $\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 168{\cdot}4$.

$\frac{B}{H} = 2{\cdot}382$. Displacement at 19 ft. 6 in. draught = about 6 610 tons. Block coefficient = about ·75. Prismatic coefficient = ·772. New value of $\frac{B}{H} = \frac{B_1}{H_1} = 2{\cdot}382 \times \frac{{\cdot}926}{{\cdot}972} = 2{\cdot}271$. Wetted surface = 23 790.

$\frac{V}{\sqrt{L}}$.	Residuary resistance in lbs. per ton of Δ, corresponding to values of B÷H.		
	2·25.	3·75.	2·271.
·65	1·3	1·968 6	1·305
·60	·955	1·46	·96
·57	...	...	·82
·70	1·895	2·853	1·907

$$
\begin{aligned}
&(10\tfrac{1}{2}\text{ knots})\quad {\cdot}82 \times 6\,610 = 5\,430 \text{ lbs. residuary resistance.}\\
&\text{Residuary H.P.} = 5\,430 \times 10{\cdot}5 \times {\cdot}003\,070\,7 = 175\\
&\text{Skin H.P.} = 21{\cdot}8 \times 23{\cdot}79 \qquad = 518\\
&\text{E.H.P.} = 693
\end{aligned}
$$

Adding 4 per cent. for appendage resistance, the E.H.P. = 720. Adding 200 air H.P., we have gross E.H.P. = 920. Taking engine efficiency = ·835, hull efficiency = 1·00, and propeller efficiency = ·58,

$$\frac{920}{{\cdot}835 \times 1{\cdot}00 \times {\cdot}58} = 1\,900 \text{ I.H.P.}$$

It will be noted that the propulsive efficiency taken from the $\frac{\text{E.H.P. (naked)}}{\text{I.H.P.}}$ = only ·365, but using gross E.H.P. we have ·48.

S.S. ——. 400·4 × 50·1 × 23 ft. mean draught. $\Delta = 8\,560$ tons. Block coefficient at 22 ft. 6 in. = ·678. Mid-area coefficient = ·960. Prismatic coefficient $= \frac{\cdot 678}{\cdot 960} = \cdot 706$. 14 knots.

4 100 I.H.P. $\frac{V}{\sqrt{L}} = \cdot 70$. $\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 133\cdot 9$. $\frac{B}{H} = \frac{50\cdot 1}{22} = 2\cdot 278$.

New $\frac{B}{H} = \frac{B_1}{H_1} = 2\cdot 278 = \frac{\cdot 926}{\cdot 960} = 2\cdot 198$. Wetted surface = 28 900 sq. ft. at 22 ft. 6 in. draught. Area exposed to air resistance roughly = 2 100 sq. ft. Suppose ship to be going against an average head wind of 18 knots, then 18 + 14 = 32 knots against the ship. For an average wind of 10 knots the air against the wind is 24 knots.

$\frac{V}{\sqrt{L}}$.	Residuary resistance in lbs. per ton of Δ, corresponding to values of B÷H.		
	2·25.	3·75.	2·198.
·70	1·341	1·9	1·321 7

$$\begin{aligned}
\text{Residuary resistance in lbs.} &= 8\,560 \times 1\cdot 321\,7 = 11\,310. \\
\text{Residuary H.P.} &= 11\,310 \times 14 \times \cdot 003\,07 = 486. \\
\text{Skin H.P. (from Table IX)} &= 49 \times 28\cdot 900 = 1\,416. \\
\text{Air resistance} = R &= \cdot 004\,3 \times 2\,100 \times (32)^2 = 9\,250 \text{ lbs.} \\
\text{Air H.P.} &= \cdot 003\,070\,7 \times 9\,250 \times 14 = 399.
\end{aligned}$$

Take engine efficiency = ·84, propeller efficiency = ·625, and allow 4 per cent. for appendage resistance, and hull efficiency, say, = 1·00.

$$486 + 1\,416 = 1\,902 \text{ naked E.H.P.}$$

With appendages = 1 980 E.H.P. Gross E.H.P. with air resistance included = 1 980 + 399 = 2 379.

$$\frac{2\,379}{\cdot 84 \times \cdot 625} = 4\,520 \text{ I.H.P. against an 18-knot wind.}$$

With a 10-knot breeze against the ship, air resistance = 5 200 lbs., air H.P. = 310, I.H.P. = 4 360.

About 8·8 per cent. of this ship's I.H.P. at full speed is expended in overcoming wind resistance when steaming against an 18-knot

wind. Against a 10-knot wind the percentage would be about 7·1. The reduction of speed against the 18-knot wind would be about $\frac{3}{4}$ knot. Against the 10-knot wind the reduction of speed would be about $\frac{1}{2}$ a knot.

S.S. ——. $355 \times 49 \cdot 25 \times 23$ ft. mean draught. $\Delta = 8\,120$ tons at 21 ft. mean draught. Block coefficient = ·775. Mid-area coefficient = ·975 at this draught. Prismatic coefficient $= \frac{\cdot 775}{\cdot 975} = \cdot 795$. $10\frac{1}{2}$ knots at 2 000 I.H.P. at sea. $\frac{V}{\sqrt{L}} = \cdot 557$.

$$\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 181 \cdot 8. \quad \frac{B}{H} = \frac{49 \cdot 25}{22} = 2\,24.$$

New $\frac{B}{H} = \frac{B_1}{H_1} = 2 \cdot 24 \times \frac{\cdot 926}{\cdot 975} = 2 \cdot 16$. Wetted surface = 26 250 sq. ft. at 21 ft. draught.

$\frac{V}{\sqrt{L}}$.	Residuary resistance in lbs. per ton of Δ, corresponding to values of B÷H.		
	2·25.	3·75.	2·16.
·70	2·074	3·245	2·007
65	1·402	2·163	1·356 3
·60	1·033	1·602	·998 9
·557	...	..	·691 9

Residuary resistance = $8\,120 \times \cdot 691\,9 = 5\,620$ lbs.
Residuary H.P. = $5\,620 \times 10\frac{1}{2} \times \cdot 003\,07 = 181 \cdot 3$.
Skin H.P. (from Table IX) = $21 \cdot 8 \times 26 \cdot 250 = 572$.

Suppose the area exposed to air resistance = 2 285 sq. ft., and the ship to be going at $10\frac{1}{2}$ knots against an average wind of 20 knots,

$$10 \cdot 5 + 20 = 30 \cdot 5 \text{ knots against the ship} = V.$$
$$\text{Air resistance} = R = \cdot 004\,3 \times 2\,285 \times (30 \cdot 5)^2 = 9\,150 \text{ lbs.}$$
$$\text{Air H.P.} = \cdot 003\,070\,7 \times 915\,0 \times 10 \cdot 5 = 295.$$

Take engine efficiency = ·83, propeller efficiency = ·61, and allow 4 per cent. for appendage resistance, and hull efficiency = 1·00. $181 \cdot 3 + 572 = 784$ gross E.H.P. without appendages, and 1 079 gross E.H.P. with appendages and air H.P.

$$\frac{1\,079}{\cdot 83 \times \cdot 61} = 2\,130 \text{ I.H.P. required to drive the ship at } 10\tfrac{1}{2} \text{ knots against a 20-knot wind.}$$

With 260 air H.P. for an $18\frac{1}{4}$-knot wind, the I.H.P. would be 2 000.

Table XXVI.—Sixth Roots of Numbers (Δ).

Δ	$\Delta^{\frac{1}{6}}$	Δ	$\Delta^{\frac{1}{6}}$	Δ	$\Delta^{\frac{1}{6}}$	Δ	$\Delta^{\frac{1}{6}}$
25	1·71	1 900	3·519	5 800	4·24	13 750	4·894
50	1·92	2 000	3·549	5 900	4·25	14 000	4·91
75	2·053 3	2 100	3·578	6 000	4·262	14 250	4·924
100	2·154	2 200	3·605	6 100	4·275	14 500	4·937
125	2·235	2 300	3·632	6 200	4·285	14 750	4·951
150	2·305	2 400	3·659	6 300	4·297	15 000	4·966
175	2·366	2 500	3·684	6 400	4·308	15 250	4·979
200	2·418	2 600	3·71	6 500	4·32	15 500	4·992
225	2·465	2 700	3·731	6 600	4·33	15 750	5·005
250	2·51	2 800	3·755	6 700	4·34	16 000	5·20
275	2·549	2 900	3·777	6 800	4·35	16 250	5·033
300	2·587	3 000	3·799	6 900	4·361	16 500	5·045
325	2·62	3 100	3·819	7 000	4·373	16 750	5·057
350	2·654	3 200	3·84	7 250	4·397	17 000	5·07
375	2·684	3 300	3·859	7 500	4·424	17 250	5·083
400	2·714	3 400	3·877	7 750	4·448	17 500	5·094
425	2·74	3 500	3·897	8 000	4·472	17 750	5·106
450	2·768	3 600	3·914	8 250	4·494	18 000	5·119
475	2·791	3 700	3·931	8 500	4·517	18 250	5·13
500	2·817	3 800	3·949	8 750	4·539	18 500	5·141
550	2·86	3 900	3·965	9 000	4·561	18 750	5·153
600	2·904	4 000	3·981	9 250	4·583	19 000	5·165
650	2·941	4 100	3·999	9 500	4·602	19 250	5·177
700	2·98	4 200	4·016	9 750	4·624	19 500	5·188
750	3·014	4 300	4·031	10 000	4·642	19 750	5·199
800	3·047	4 400	4·049	10 250	4·661	20 000	5·21
850	3·079	4 500	4·063	10 500	4·68	20 500	5·231
900	3·107	4 600	4·079	10 750	4·699	21 000	5·252
950	3·136	4 700	4·092	11 000	4·718	21 500	5·273
1 000	3·17	4 800	4·107	11 250	4·735	22 000	5·292
1 100	3·211	4 900	4·12	11 500	4·752	22 500	5·312
1 150	3·236 8	5 000	4·135	11 750	4·769	23 000	5·331
1 200	3·26	5 100	4·149	12 000	4·785	23 500	5·35
1 300	3·305	5 200	4·161	12 250	4·801	24 000	5·37
1 400	3·347	5 300	4·175	12 500	4·817	24 500	5·389
1 500	3·383	5 400	4·189	12 750	4·834	25 000	5·407
1 600	3·42	5 500	4·201	13 000	4·85	25 500	5·425
1 700	3·454	5 600	4·215	13 250	4·865	26 000	..
1 800	3·487	5 700	4·228	13 500	4·88	26 500	..

TABLE XXVI.—SIXTH ROOTS OF NUMBERS (Δ)—*continued.*

Δ	$\Delta^{\frac{1}{6}}$	Δ	$\Delta^{\frac{1}{6}}$	Δ	$\Delta^{\frac{1}{6}}$	Δ	$\Delta^{\frac{1}{6}}$
27 000	..	34 500	5·706	42 000	5·894	49 500	6·06
27 500	..	35 000	5·72	42 500	5·906	50 000	6·07
28 000	5·510 5	35 500	5·734	43 000	5·918	51 000	..
28 500	5·528	36 000	5·748	43 500	5·93	52 000	..
29 000	5·543	36 500	5·76	44 000	5·94	53 000	..
29 500	5·56	37 000	5·774	44 500	5·951	54 000	..
30 000	5·574	37 500	5·787	45 000	5·961	55 000	6·166
30 500	5·59	38 000	5·80	45 500	5·972	56 000	..
31 000	5·604	38 500	5·812	46 000	5·984	57 000	..
31 500	5·62	39 000	5·824	46 500	5·994	58 000	..
32 000	5·634	39 500	5·836	47 000	6·005	59 000	..
32 500	5·649	40 000	5·849	47 500	6·017	60 000	6·257
33 000	5·663	40 500	5·86	48 000	6·028	65 000	..
33 500	5·678	41 000	5·871	48 500	6·039	70 000	6·42
34 000	5·691	41 500	5·883	49 000	6·049		

MR R. E. FROUDE'S TYPE 4, SERIES A. Δ = 6 048 tons.
K = 2·8. (See p. 76.)

M.	Length.	O.	$L^{-\cdot 175}$	$OSL^{-\cdot 175}$	$\frac{OSL^{-\cdot 175}}{C} \times$ E.H.P. = Froude's Skin H.P.	$\frac{\text{Froude's Skin H.P.}}{\text{Tideman's Skin H.P.}}$
4·6	274	·077 4	·954	·415	2 544	·865 *
5·0	298	·076 49	·961	·431 5	2 900	·948
5·453	325	·076 17	·968 2	·451 9	3 030	·956
6·0	358	·075 2	·976	·473	3 166	·955
6·6	393·5	·074 2	·984 8	·493 6	3 304	·955
7·0	418	·073 76	·990	·508	3 400	·957
7·4	441	·073 29	·995	·520	3 480	·956

$$\text{E.H.P.} - \text{skin H.P} = \text{Residuary H.P.,}$$

$$\text{Residuary resistance lbs. per ton } \Delta = \frac{\text{Residuary H.P.}}{V \times \cdot 003\ 07 \times 6\ 048},$$

* *I.e.* except for the two abnormally short vessels, Froude's skin H.P. works out about 4½ per cent. less than the skin H.P. from our Table IX, based upon Tideman's constants. The skin frictional H.P. by Froude is $V^{2\cdot 825}$ × Froude's surface friction constants.

while the skin H.P. by our tables is $V^{2\cdot 83} \times$ Tideman's surface friction constants.

Rear-Admiral Taylor uses the latter, and from this constructs his fig. 78, a diagram showing contours of skin frictional resistance in lbs. per ton Δ, for a ship with wetted surface from 4 to 7 per cent. below the average.

Total resistance of model—Tideman's skin resistance = Taylor's residuary resistance . . . . . . A

Total resistance of model—Froude's skin resistance = residuary resistance according to Froude and Baker . . . . B

When using A, it should be remembered that the values of residuary resistance per ton Δ are lower by the $4\frac{1}{2}$ per cent. or so, mentioned above, than in the case of B.

In design work, then, when powering a ship, if we have no information as to total resistance or E.H.P. from tank trial, or S.H.P. from an exactly similar vessel, and if we have no Ⓒ values, if $\dfrac{\Delta}{\left(\frac{L}{100}\right)^3}$ is not over 160, we may use A, adding, say, 5 per cent. to the residuary resistance per ton Δ so found, to bring it into line with Froude.

Then for skin H.P., if, in the case we are dealing with, $\dfrac{\Delta^{\frac{2}{3}}}{\left(\frac{L}{100}\right)^3}$ does not exceed 160, we may use Taylor's fig. 78, adding a percentage to the reading equivalent to the difference between 15·4 and our value of C in Taylor's formula for wetted surface, using Taylor's fig. 41 to find the value of C. In many ordinary vessels, C = 16·5, *i.e.* 7 per cent. in excess of the 15·4 upon which Taylor's fig. 78 is based.

For both residuary resistance and skin resistance the above values are given for naked models, *i.e.* models without appendages.

Appendage resistance is largely eddy-making, and should be added to the residuary resistance, say 4 per cent. for single-screw ships. For twin-screw ships with shaft bossings not too favourably arranged, the additional resistance to add for appendages may be anything between 10 and 20 per cent.

The extra-wetted surface of the appendages is another matter. This may be added to the wetted surface of the naked ship, and additional skin friction allowed, considerably less perhaps, how-

ever, than the proportional increase of surface causing it, as this part of the surface is said to carry a body of water with it.

Thus if we take Taylor's wetted surface, we have $C\sqrt{DL} \times f \times V^{2\cdot83}$ = skin friction H.P. of naked hull. The percentage of surface to add for appendages is similarly multiplied by $V^{2\cdot83} \times$ a fraction of f, and the total gives the skin H.P.

Or, if we take Mumford's formula for wetted surface, we have $(L \times D \times 1\cdot7) + (L \times B \times \text{block coefficient}) \times f \times V^{2\cdot83}$ = skin H.P. of naked hull, unless, as is often the case, we add or deduct something to Mumford's product to correct it for the particular type of hull in question. Appendage surface friction is then added as before.

Deduction of Froude's Surface Friction Coefficients from Mr G. S. Baker's 1913 Models.

Values of Ⓒ scaled from diagrams. Taking Set C, and working back to find f the coefficient of fluid friction. Model 18A. Wetted surface S = 30 860 sq. ft., S = 6·39. Wetted surface taken as 2 per cent. above Mumford's, and as 4·4 per cent. below Taylor's, wetted surface.

(1) At $\dfrac{V}{\sqrt{L}} = \cdot711.$ 14·22 knots for 400-ft. ship.

Ⓒ = ·732.

Ⓛ = ·750 5.

$OSL^{-\cdot175} = \cdot521.$ Skin H.P. = 1 582.

Let us find the value of f in the formula

$$\text{Skin H.P.} = f \,.\, S \times \cdot003\,070\,7 \times V^{2\cdot825}$$
$$(14\cdot22)^{2\cdot825} = 1\,803.$$

$$\therefore\ f = \frac{1\,582}{30\,860 \times \cdot003\,070\,7 \times 1\,803} = \cdot009\,26.$$

(2) At $\dfrac{V}{\sqrt{L}} = \cdot633.$ 12·66 knots for 400-ft. ship.

Ⓒ = ·763.

Ⓛ = ·669.

$OSL^{-\cdot175} = \cdot532.$ Skin H.P. = 1 139.

$(12\cdot66)^{2\cdot825} = 1\,300.$

$$\therefore \quad f = \frac{1\,139}{30\,860 \times \cdot 003\,070\,7 \times 1\,300} = \cdot 009\,245.$$

These values of f are about 5 per cent. higher than the standard value of Froude's f, viz. ·008 83, quoted by Mr G. S. Baker. Perhaps it would be better to write skin H.P. $= (\cdot 008\,83 \times S \times \cdot 003\,070\,7 \times V^{2\cdot 825})1\cdot 05$, to show that Mr Baker has added the 5 per cent. for form (see pp. 5, 6, and 34).

Example.—R. E. Froude, 1904. Series A, Type 4.

2·8 = the speed constant Ⓚ

$$\text{Ⓚ} = \frac{V}{\Delta^{\frac{1}{6}}} \times \cdot 583\,4 \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$\therefore \quad V = \frac{\Delta^{\frac{1}{6}} \times 2\cdot 8}{\cdot 583\,4} = \frac{(6\,048)^{\frac{1}{6}} \times 2\cdot 8}{\cdot 583\,4}.$$

$$\log(6\,048)^{\frac{1}{6}} = \tfrac{1}{6}\log 6\,048 = \tfrac{1}{6} \times 3\cdot 781\,62 = \cdot 630\,27$$
$$= \log 4\cdot 268.$$
$$\therefore \quad (6\,048)^{\frac{1}{6}} = 4\cdot 268.$$
$$\therefore \quad V = \frac{4\cdot 268 \times 2\cdot 8}{\cdot 583\,4} = 20\cdot 5.$$

The resistance constant

$$\text{Ⓒ} = \frac{\text{E.H.P.}}{\Delta^{\frac{2}{3}} V^3} \times 427\cdot 1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$\therefore \quad \text{E.H.P.} = \frac{\text{Ⓒ} \times (6\,048)^{\frac{2}{3}} \times (20\cdot 5)^3}{427\cdot 1}$$

$$= \frac{\cdot 962 \times 332 \times 8\,615}{427\cdot 1} = 6\,450.$$

The length constant

$$\text{Ⓜ} = \frac{L}{\Delta^{\frac{1}{3}}} \times \cdot 305\,7 \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If

$$\text{Ⓜ} = 5\cdot 453,$$

$$\therefore \quad L \times \frac{\cdot 305\,7}{\Delta^{\frac{1}{3}}} = 5\cdot 453 \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$\therefore \quad L = \frac{5\cdot 453 \times 18\cdot 22}{\cdot 305\,7} = 325.$$

$$B = \frac{\cdot 956 \times 18\cdot 22}{\cdot 305\,7} = 57.$$

$$\text{(C) for 300-ft. correction} = \begin{array}{r} \cdot 965 \\ \cdot 003 \\ \hline \cdot 962 \end{array}$$

$$D = 22.$$

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = \frac{332 \times 8\,615}{12\,900} = 222 \quad \text{if} \quad \frac{\text{E.H.P.}}{\text{I.H.P.}} = \cdot 50.$$

Take wetted surface = 21 780. [Mumford's formula gives 21 810.]

$$\text{(S)} = \frac{21\,780}{332} \times \cdot 093\,46 = 6 \cdot 13.$$

Example.—R. E. Froude, I.N.A., 1904. Series A, Type 4.

(K) = 2·8. (M) = 7·4. (C) = ·76. Let $\Delta = 6\,048$. $V = 20 \cdot 5$.

$$V = \frac{K \times \Delta^{\frac{1}{6}}}{\cdot 583\,4} = \frac{2 \cdot 8 \times 4 \cdot 268}{\cdot 583\,4} = 20 \cdot 5. \quad \frac{\text{Beam}}{\text{Draught}} = \frac{57}{22}.$$

$$\begin{array}{r} \cdot 782\,4 \\ \cdot 760\,0 \\ \hline \cdot 022\,4 \\ \hline \end{array}$$

$$\text{Corrected value of (C)} = \cdot 737\,6$$

$$\text{E.H.P.} = \frac{\cdot 737\,6 \times 332 \times 861\,5}{427 \cdot 1} = 494\,0.$$

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = \frac{332 \times 861\,5}{988\,0} = 290 \quad \text{when} \quad \frac{\text{E.H.P.}}{\text{I.H.P.}} = \cdot 50.$$

The length-speed constant

$$\text{(L)} = \frac{K}{\sqrt{M}} = \frac{2 \cdot 8}{\sqrt{7 \cdot 4}} = \frac{2 \cdot 8}{2 \cdot 645} = 1 \cdot 059 \quad . \quad . \quad . \quad (3)$$

Length of ship

$$L = \frac{7 \cdot 4 \times 18 \cdot 22}{\cdot 305\,7} = 441.$$

$$\text{(B)} = \cdot 824 = \frac{\text{Beam}}{\Delta^{\frac{1}{3}}} \times \cdot 305\,7.$$

$$\therefore \quad \text{Beam} \times \frac{\cdot 305\,7}{\Delta^{\frac{1}{3}}} = \cdot 824. \quad \tfrac{1}{2}B = \cdot 410\,6.$$

$$\therefore \quad \text{Beam} = \cdot 821\,2 \times \frac{\Delta^{\frac{1}{3}}}{\cdot 305\,7} = \frac{\cdot 821\,2 \times 18 \cdot 22}{\cdot 305\,7} = 48 \cdot 6.$$

$$\text{(D)} = \cdot 315. \quad \text{Draught} = \frac{\cdot 315 \times 18 \cdot 22}{\cdot 305\,7} = 18 \cdot 78.$$

Dimensions: $441 \times 48 \cdot 6 \times 28 \cdot 78$, $w = \cdot 525$, $\Delta = 604\,8$.

Take wetted surface $S = 25\,360$.

Then

$$\text{(S)} = \frac{25\,360}{332} \times \cdot 093\,46 = 7 \cdot 14$$

$$\text{OSL}^{-\cdot 175} = \cdot 073\,29 \times 7 \cdot 14 \times \cdot 990\,2.$$

$$= \cdot 518.$$

CHAPTER VIII.

PROPELLERS.

A SCREW propeller impels a column of water in a sternward direction. Suppose the propeller to be working so far behind the ship that it is not in the wake or following current, then the speed of the column of water driven aft or pumped aft by the screw, *i.e.* the speed of the propeller race, is $V_A - V_S$ = the slip, or real slip, *i.e.* a speed given to the water acted upon by the propeller and driven sternwards, where $V_A = pN$ = pitch in feet $\times$ revolutions per minute in this case, and V_S = speed of ship in feet per minute. If the screw is advancing into undisturbed water, in the manner above described, it is developing a certain thrust T, required to drive the ship. Then the propeller efficiency (e_2) under these conditions would be the ratio $\frac{\text{E.H.P.}}{\text{D.H.P.}}$, where E.H.P. is the power corresponding to the net or tow-rope resistance of the ship, and D.H.P. the delivered horse-power or power delivered to the propeller, D.H.P. = S.H.P. less the power lost in friction of the stern tube and its packing, or = I.H.P. less the power lost by friction of engines, dependent pumps, shafting, thrust-block, and stern tube. But the propeller, instead of working in undisturbed water, works in the wake or current of water following the ship, and instead of meeting the water at a speed equal to the ship's speed, it is caused to advance through the water around it at a speed = the ship's speed minus the speed of the wake, *i.e.* $V_S - wV_S = V_A$ = "the speed of advance." The thrust horse-power = TV_A. The useful work, so far as the ship is concerned, is always TV_S, whether the propeller is working in undisturbed water far behind the ship or working in the wake water in its usual position at the stern of the ship.

If the propeller imparts movement to a column of water asternwards, the reaction of the water produces the thrust. If there is no wake, *i.e.* if the propeller is working in undisturbed or "open water," the speed with which the propeller meets the water is

simply the speed of advance of the propeller, and the difference between $(P \times N)$ and the speed of advance = the real slip; but if the propeller is working in its usual position at the stern of a vessel going ahead, the propeller meets water which already has a forward motion, and has to destroy this forward motion of the water and impress a real sternward motion upon it. It does this gradually, and the acceleration commences in front of and before the water reaches the propeller, by a kind of suction towards the back of the blade. In the latter case the difference between $(P \times N)$ and the speed of advance is the apparent slip. In an experimental basin, when the propeller is mechanically caused to advance through open water at the speed $(P \times N)$ feet per minute, there is no slip and no thrust. *The wake has not a single uniform speed, but has different speeds at different parts of the stern and at different levels.* The wake is practically the same (though perhaps not exactly) on the port side of a propeller as it is on the starboard side. The actual wake, however, is considered as sensibly equivalent to a uniform wake, and the slip is mean slip. In the same ship the wake speed with inward-turning screws is different from the wake with outward-turning screws. Sir A. Denny, Bart., mentions the case of a twin-screw yacht in which the wake was 11 per cent. for inward and 17½ per cent. for outward turning; hull efficiency inward ·95, and outward 1·03. The mean real slip, then, is greater than the apparent slip by the amount of this wake. The wake fraction or wake speed is equal to the real slip ratio or real speed minus the apparent slip ratio or speed.

If A = the cross-sectional area of the race or column of water projected sternwards, in square feet,
W = weight of a cubic foot of sea water in lbs.,
v = speed of race, in feet per second, relatively to the ship,
V_S = speed of ship in feet per second,
m = mass of water acted on by the propeller per second, in lbs.,

$$m = \frac{W}{g} A \times v,$$

the momentum of the race $= \frac{W}{g} A \times v(v - V_S)$, and this is the measure of the thrust of the screw, T, to overcome the resistance of the ship augmented by the wind, waves, pitching, appendages, and the effect of the presence of the ship upon the propeller (wake effect) and the effect of the presence of the propeller upon the ship (augmentation of the ship resistance by the defect of pressure behind the stern, due to the action of the propeller in sucking the water forward of itself, the beginning of the accelera-

tion imparted to the water). If we say the race has an absolute velocity aft of u feet per second, then $T = \frac{W}{g}A(V_S+u)u$.

The useful work of the propeller is $TV_S = \frac{W}{g}A(V_S+u)V_S u$.

The speed of advance of the propeller through the water in which it works is usually less than the speed of the ship. Propellers usually advance a distance less than their pitch for each revolution.

If V_A = the speed of advance of the propeller through the wake water,
and V_S = speed of ship,
wV_S = speed of wake,

$$V_A = V_S - wV_S = V_S(1-w).$$

If the propeller is made to advance at a speed which will give no slip, viz. $P \times N$(P = pitch, N = revolutions), $P \times N$ is called the speed of the propeller. $(P \times N) - V_A$ = the speed of the slip.

$$\frac{\text{Speed of propeller} - \text{speed of advance}}{\text{Speed of propeller}} = \text{real slip ratio} = S.$$

$$\frac{\text{Speed of propeller} - \text{speed of ship}}{\text{Speed of propeller}} = \text{apparent slip ratio} = S_1.$$

When (as nearly always) the speed of advance V_A is less than the speed of the ship V_S, real slip ratio is greater than apparent slip. Pitch × revolutions, P.R., is termed the speed for no slip, or the speed of the propeller. If the propeller advanced a distance = P each revolution, there would be no slip. The slip is $P \times s$, the propeller advances $P-(P\times s)$. It advances $P(1-s)$ each revolution, and the speed of advance is $P(1-s)R$. The useful work $\sim T \times P(1-s)R$ per minute. The gross work, or work delivered to the screw [corresponding to the D.H.P. (where horse-power delivered to propeller = $e_1 \times$ I.H.P.)], is the torque $\times 2\pi R$, R being revolutions per minute.

Let Q = torque, T = thrust, as before.

$$\text{Efficiency} = \frac{\text{Useful work}}{\text{Gross work}} = \frac{T \times P(1-s)R}{2\pi \,.\, QR} = \frac{T}{Q} \times \frac{P(1-s)}{2\pi}.$$

The results of Mr Taylor's model experiments upon propellers are plotted as curves of thrust in lbs., torque in pound-feet, and efficiency, upon real slip ratio as abscissæ. There is a difference between nominal pitch (the pitch of the driving face of the

blade) and virtual pitch (the effective pitch as modified by the curved back of the blade), which causes some thrust to be registered at the speed for zero slip, P × R, and both Mr Taylor's curves and Mr R. E. Froude's 1908 curves allow for this. Prof. T. B. Abell's analysis (*Trans. Inst. N.A.*, 1910) makes this very clear, showing that the pitch for no thrust is not always the same for a given propeller, but seems to change with the speed of advance.

Sir A. Denny's address to the Institution of Marine Engineers, 1915, confirms this (see p. 166). The uncertainty of pitch makes all propeller calculations based upon present information rather unsatisfactory.* Mr H. Gibson has measured thrust in tons by meter.

Owing to the wake, the thrust of the propeller is greater than it would be if it were working in still water. Part of the work of the machinery propelling the ship and causing wake is returned as useful work in the form of an addition to the thrust. This is usually called "the gain due to wake," or "the wake gain." It is less in twin-screw ships than in single-screw ships. This gain is practically balanced by the thrust deduction which is due to the reduction of water-pressure behind the ship (equivalent to an augmentation of the resistance against which the ship moves), caused, as previously stated, by the sucking action of the screw upon the water just forward of the blades. Some distance forward of the screw the water is sucked aft towards the blades, which impart a gradually increasing acceleration to it when driving it sternwards. The fore-and-aft position of the screw on the ship affects both the wake gain and the thrust deduction, causing the ship to act more or less upon the propeller by the wake, and the screw to produce more or less suction upon the ship according to its situation.

If T = the thrust, and R = the net or tow-rope resistance of the ship, T − R = thrust deduction.

If tT is the fractional amount by which T exceeds R, t being the thrust deduction coefficient,

$$R = T(1-t).$$
$$(1-t) = \text{the thrust deduction factor.}$$
$$(1-t) = \frac{R}{T}.$$

* A case was mentioned in which model propellers were driven along a tank, with no ship model in front of them, at a speed of 500 ft. per min. With slip ratio of zero, *i.e.* when pN = 500 ft. per min., it was expected that no thrust would be registered, but this was not the case, for at zero thrust the pitch actually is ·588, not ·50. Probably the difference of pitch is greater when the symmetrical ogival section is departed from and a blade having its greatest thickness, say ⅓, from the leading edge is used.

WAKE.

Since $V_A = V_S(1-w)$, $\therefore \dfrac{V_S}{V_A} = \dfrac{1}{(1-w)}$ = the "wake factor," where w = the "wake fraction."

$$\frac{V_S}{1+w_p} = V_A.$$

$$V_A = V_S - wV_S$$

$$= V_S(1-w).$$

$$\therefore \frac{1}{1+w_p} = (1-w).$$

$$1+w_p = \frac{1}{1-w}.$$

$$\therefore w_p = \frac{1}{1-w} - 1 \qquad (1)$$

$$w = \frac{w_p}{1+w_p} \qquad (2)$$

where V_S = speed of ship in knots.
V_A = speed of advance of propeller in knots through the wake water.
w = Taylor's wake fraction.
w_p = Froude's wake percentage.

The speed of advance V_A is the same, whether we calculate it from w_p or from w.
For single screws, $w = -\cdot05 + (\cdot5 \times b)$.
For twin screws, $w = -\cdot2 + (\cdot55 \times b)$.

Froude's method of propeller design works upward from E.H.P., and the first step in using this method is to multiply E.H.P. by a factor greater than unity, to allow for wind, rough water, pitching, appendage resistance, etc., to arrive at the T.H.P. Taylor's method works directly downward from S.H.P., which includes propeller efficiency. The choice of a method depends upon the data at the command of the estimator. Taylor's based upon D.H.P. would be even better, and, of course, D.H.P. can be used instead of S.H.P., the shaft transmission efficiency shown on Messrs M'Laren and Welsh's diagram being the only difference between S.H.P. and D.H.P.*

Both of the above-named systems are based upon the most elaborate tank experiments which have been carried out up to the present time.

* *Trans. Inst. Engineers and Shipbuilders, Scot.*, 1914-15.

Further experiments are required to compare experimental data from model propellers with results of corresponding full-sized screws.

The E.H.P. $\sim RV_S$, and E.H.P. $= R \times V_S \times {}^{\cdot}003\,07$.

The T.H.P. of the screw $\sim TV_A$, and T.H.P. $= T \times V_A \times {}^{\cdot}003\,07$.

$$e_3 = \frac{\text{E.H.P.}}{\text{T.H.P.}} = \text{hull efficiency} = \frac{RV_S}{TV_A} = \frac{1-t}{1-w}.$$

$$= \frac{R}{T} \times \frac{V_S}{V_A} = (1-t) \times \frac{1}{(1-w)}.$$

If the screw be set to work in still water apart from the ship, or driven along the tank in the open without any model in front of it, at same revolutions as when attached to ship, but its speed of advance adjusted so that the same thrust is obtained from the screw as when it worked in its usual position on the ship, and if S_1 be the power delivered to the propeller under these conditions, then $\frac{TV_A}{S_1}$ is the screw efficiency in still water (as in model experiments), the ratio of the work got out to the work put in, and the propulsive efficiency of the screw

$$e_2 = \frac{TV_1}{S_1} \times (\text{relative rotative efficiency}).$$

The relative rotative efficiency is not always taken into account. It is the ratio

$$\frac{S_1}{\text{D.H.P.}},$$

or ratio

$$\frac{\text{The power delivered to the propeller for developing a certain thrust in open water.}}{\text{The power delivered to the propeller for developing same thrust when working in the water behind the ship.}}$$

As stated on p. 6, the propulsive efficiency of the ship $\rho = e_1 \times e_2 \times e_3$; $\rho =$ engine efficiency $\times$ screw efficiency $\times$ hull efficiency

$$\rho = \frac{\text{D.H.P.}}{\text{I.H.P.}} \times e_2 \times \frac{\text{E.H.P.}}{\text{T.H.P.}}$$

Mr Luke's paper in 1910 to the Institution of Naval Architects gave values of relative rotative efficiency and hull efficiency obtained from analyses by Mr Froude and Signor Pecoraro. These we have tabulated on p. 149.

TABLE XXVII.—WAKE AND THRUST DEDUCTION, from Mr Luke's 1910 paper quoting Mr R. E. Froude's 1898 figures for the forms given in the paper. (Inward-turning and outward-turning screws.) See Table XXV.

Ship.		Wake.		Thrust deduction.		Hull efficiency.		Relative rotative efficiency.	
		Outward turning.	Inward turning.	Out.	In.	Out.	In.	Out.	In.
Battleships	1	·165	·168	·185	·175	·948	·965	·999	·999
	2	·095	·105	·095	·105	·990	·989	·992	1·003
	3	·092	·098	·098	·100	·985	·987	·999	1·005
Cruisers	4	·095	·107	·095	·098	·990	·997	·990	1·010
	5	·075	·090	·083	·090	·985	·992	·990	1·010
	6	·095	·108	·103	·110	·982	·985	·997	1·007
	7	·082	·090	·101	·105	·973	·975	·992	1·007
	8	·087	·092	·115	·135	·961	·945	·987	1·013
	9	·087	·098	·100	·100	·978	·987	·999	1·005
	10	·085	·098	·099	·100	·976	·987	·999	1·004
	11	·060	·067	·075	·080	·980	·982	·990	1·008
	12	·082	·084	·100	·103	·975	·972	·987	1·005
	13	·040	·045	·065	·070	·972	·972	·987	1·005
	14	·040	·050	·065	·068	·972	·978	·990	1 004
	15	·068	·074	·088	·096	·972	·970	·992	1·008
Extreme shallow draught vessel	16	·160	·155	·12	·113	1·021	1·025	1·002	1·008
Torpedo-boat destroyers	17	·042	·040	·040	·035	1·000	1·002	1·015	1·009
	18	·012	·016	·037	·033	·974	·982	·985	1·002
	19	– ·010	– ·008	·020	·015	·970	·975	·987	·998
	20	– ·007	– ·001	·015	·015	·975	·983	·985	1·005
	21	– ·015	– ·010	·016	·016	·970	·974	·995	1·003

In Froude's nomenclature w the "wake fraction" differs from Taylor's "wake fraction."

Froude's is $\dfrac{\text{Wake velocity}}{\text{Speed of propeller in “open water”}}$.

Taylor's "wake fraction" is a fraction of the ship's speed.
MacDermott's "wake factor" is also a fraction of the ship's speed.

Some further explanation is required of the terms "wake factor," "wake fraction," "wake percentage," "speed of the wake." In

the above we have used Mr Taylor's expressions, not Mr Froude's, for these values.

The formulæ for wake give the value of w, the "wake fraction." Thus Mr Taylor's $w = -\cdot 2 + \cdot 55b = t$ roughly shows the "wake fraction" and the thrust deduction "coefficient" equal, as they nearly are in most cases of twin screws. When this is the case, the "wake factor" $\frac{1}{1-w}$ and the "thrust-deduction factor" $(1-t)$ are reciprocals.

Mr Taylor and others express the wake as a fraction of the ship speed V_S; thus wV_S = actual speed of wake where w = the "wake fraction" and $\frac{V_S}{V_A} = \frac{1}{1-w}$ = the "wake factor."

Mr Froude's nomenclature, used also by Mr G. S. Baker and Mr W. J. Luke, is different in that $\frac{V_S}{V_A} = 1 + w_p$ where w_p is the "wake percentage," the wake being expressed as a fraction of V_A, the speed of advance of the screw, and the wake factor is $1 + w_p$.

Mr Taylor's w = Mr Froude's $\frac{w_p}{1+w_p}$.

Mr Taylor's "wake factor" $= \frac{1}{1-w}$, and Mr Froude's "wake factor" $= 1 + w_p$, but these are equal. The wake factor is greater than unity.

Example.—Let the speed of the ship V be 20 knots, and let the "wake percentage" according to Froude and Baker be $w_p = \cdot 16$. Then, according to Froude, the speed of the screw V_A through the wake water

$$V_A = \frac{V_S}{1+w_p} = \frac{20}{1\cdot 16} = 17\cdot 24 \text{ knots.}$$

$$1 + w_p = 1\cdot 16 = \text{"wake factor."}$$

Now Taylor's "wake factor" is also $1\cdot 16 = \frac{1}{1-w}$ where $w = \frac{w_p}{1+w_p} = \cdot 138$.

Taylor's formula is $V_A = V_S - wV_S$, which gives $V_A = 17\cdot 24$ knots.

Taylor calls w the "wake fraction."

In both systems $\frac{V_S}{V_A}$ = the "wake factor."

The thrust-deduction factor is less than unity. They do not

necessarily quite balance one another, and they are both factors in the efficiency.

The wake percentage is slightly higher with models than with full-sized ships.

OTHER FORMULÆ FOR WAKE.

(1) In the discussion on three papers in 1910 read before the Institution of Naval Architects, Mr P. A. Hillhouse gave a useful expression for wake, deduced from Professor M'Dermott's figures:—

$$W = 30p - \cdot 75\frac{L}{B} - \cdot 55,$$

where W = wake percentage (a percentage of the speed of the ship).
p = prismatic coefficient.
L = length on water-line.
B = breadth on water-line.

As the forward motion is gradually impressed on the water as the vessel moves through it, the speed of the wake is greater in the case of long vessels.

(2) Mr D. W. Taylor, in the same discussion, gave a formula for w, the wake fraction, the ratio between the speed of the wake and the speed of the ship V_S (not V_A, as in Froude's wake, see p. 150), in terms of block coefficient b. For twin screws

$$w = -\cdot 2 + \cdot 535b,$$

and thrust deduction

$$t = -\cdot 198 + \cdot 557b.$$

Nearly identical expressions, confirming a former dictum of Mr R. E. Froude, that for twin-screw vessels, on the average, wake factor and thrust deduction neutralise each other and hull efficiency is unity. From this he suggested the following single approximate formula,

$$w = -\cdot 2 + \cdot 55b = t,$$

for cases of twin screws in which shaft bossing does not materially modify the natural flow of the water.

(3) For twin-screw steamers the lower line on Plate 66 answers fairly well, the equation being

$$w = -\cdot 155 + \cdot 44b.$$

OTHER FORMULÆ FOR WAKE VALUES.

(4) Mr D. W. Taylor's *Resistance of Ships and Screw Propulsion*, published twenty years ago, gave equations for mean wake factor, or wake coefficient, a fraction of the speed of the ship, as follows :—

$$\text{For single screws, } w = 0{\cdot}44\omega - 0{\cdot}02,$$
$$\text{For twin screws, } w = 0{\cdot}57\omega - 0{\cdot}20,$$

where w = wake factor (wV), V being the speed of ship,
and ω = block coefficient.

These values were used by Dr Caird for the analysis of the trials of the Dutch opium cruiser "Argus" (Plate 35). A mean line through Mr Froude's values, quoted by Mr Luke before the Institution of Naval Architects in 1910, for twin screws, falls somewhat lower, and has the equation suggested in 1910 by Mr Taylor, and already mentioned, viz. :

$$w = -{\cdot}2 + {\cdot}535b,$$

where b = block coefficient.

(5) Another formula, of the form suggested by Taylor, was given in an article on "Screw Propellers" in *The Shipbuilder*, September 1913, by Mr A. J. C. Robertson :—

For single-screw ships, wake $= -{\cdot}05 + {\cdot}45 \times$ prismatic coefficient,
For twin-screw ships, wake $= -{\cdot}20 + {\cdot}50 \times$ prismatic coefficient.

Having regard to the comparatively high prismatic coefficients of torpedo craft, their low block coefficients, and their small ratio of length to beam, on the whole we lean to Mr Taylor's $w = -{\cdot}2 + {\cdot}535b$ for twin screws, or Mr Hillhouse's $W = 30p - {\cdot}75\frac{L}{B} - 5{\cdot}5$.

WAKE.

In a paper read before the American Society of Naval Architects and Marine Engineers in 1896, Professor MacDermott gave a good formula for the speed of the wake, as a percentage of the speed of the ship, applicable to both twin screws and single screws.

L = length of vessel in feet measured from fore side of stem to after side of inner stern-post.
p = prismatic coefficient.
m = midship-area coefficient.

SINGLE-SCREW VESSELS.

Formula $w = 0{\cdot}16\left(\frac{p}{m}L^{\frac{1}{3}} - 0{\cdot}6\right)$. Wake percentage = $100w$.

(From Professor MacDermott's paper.)

Name.	Length.	Prismatic coefficient.	Mid-area coefficient.	Actual wake per cent.	Computed wake per cent.
Flavio Gioja . .	249	·619	·85	19·36	19·63
Charles V . .	306	·658	approx. ·93	19·57	19·79
Albacore . .	128	·722	·825	23·66	22·82
Gallia . . .	419	·711	·92	25·07	24·21
Servia . . .	503	·723	·91	25·07	26·26
City of Rome .	534	·713	·925	25·07	25·54
Warrior (old) .	367	·671	·825	25·44	25·22
Great Eastern .	666	·61	·82	25·44	25·57
Cumus . . .	218				27·49
Encounter . .	213	·68	·75	26·29	27·36
Opal . . .	213				27·36
A. . . .	158	·612	approx. ·82	17·39	18·18

TWIN-SCREW VESSELS.

$w = 0{\cdot}13\left(\frac{p}{m}L^{\frac{1}{3}} - 1{\cdot}1\right)$. Wake percentage = $100w$.

(From Professor MacDermott's paper.)

Name.	Length.	Prismatic coefficient.	Mid-area coefficient.	Actual wake per cent.	Computed wake per cent.
Surprise . .	250	·535	·858	5·46	6·05
Iris . . .	300	·548	·909	5·46	5·98
Orlando Class .	300	·563	·879	8·28	7·25
Admiral Class .	325	·656	·857	10·72	11·79
Italia . . .	400	·655	·867	12·28	12·35
Conqueror . .	270	·702	·851	13·87	12·96
Great Eastern .	666	·61	·825	14·45	14·01
Devastation . .	285	·767	·888	15·04	14·51
Dnilo . . .	340	·775	·874	15·94	16·16
A. . . .	158	·612	·82	8·54	8·27
B. . . .	257	·75	·875	14·9	13·8

In Mr Baker's book, *Ship Form, Resistance and Screw Propulsion*, the wake factor is given for typical vessels all brought to a standard length of 400 feet. The wake factor is the same as Mr R. E. Froude's wake percentage. If we call it w_p, and Taylor's wake fraction w, we have

$$w = \frac{w_p}{1 + w_p}.$$

$$\text{If} \quad w_p = \cdot 20, \qquad w = \frac{\cdot 20}{1 + \cdot 20} = \cdot 166.$$

$$\text{If} \quad w_p = \cdot 15, \qquad w = \frac{\cdot 15}{1 + \cdot 15} = \cdot 130\,5.$$

$$\text{If} \quad w_p = \cdot 33, \qquad w = \frac{\cdot 33}{1 + \cdot 33} = \cdot 248.$$

$$\text{If} \quad w_p = \cdot 04, \qquad w = \frac{\cdot 04}{1 + \cdot 04} = \cdot 038\,5.$$

MacDermott's Formula for Wake Fraction.

(1) S.S. ——. $400 \cdot 4 \times 50 \cdot 1 \times 23 \cdot 5$ ft. mean draught. Block coefficient = ·68. Mid-area coefficient = ·961. Prismatic coefficient = ·708. $\frac{p}{m} = \frac{\cdot 708}{\cdot 961} = \cdot 736$. $L^{\frac{1}{6}} = (400 \cdot 4)^{\frac{1}{6}} = 2 \cdot 714$.

$$\begin{aligned} w &= 0 \cdot 16[\cdot 736 \times 2 \cdot 714 - (0 \cdot 6)] \\ &= 0 \cdot 16 \times (2 - \cdot 6) \\ &= 0 \cdot 16 \times 1 \cdot 4 = \cdot 224. \end{aligned}$$

(2) S.S. ——. $375 \cdot 2 \times 47 \cdot 8 \times 23 \cdot 5$ ft. mean draught. $\Delta = 7\,654$. Block coefficient = ·636. Mid-area coefficient = ·966. Prismatic coefficient = ·659. $\frac{p}{m} = \cdot 681$. $L^{\frac{1}{6}} = (375 \cdot 2)^{\frac{1}{6}} = 2 \cdot 684$.

$$\begin{aligned} w &= \cdot 16(\cdot 681 \times 2 \cdot 684 - \cdot 6) \\ &= \cdot 16(1 \cdot 83 - \cdot 6) = \cdot 16 \times 1 \cdot 26 \\ &= \cdot 202. \end{aligned}$$

(3) S.S. ——. $340 \times 46 \cdot 5 \times 23 \cdot 33$ ft. mean draught. 8 000 tons displacement. Block coefficient = ·76. Mid-section coefficient = ·975. Prismatic coefficient = ·78. $\frac{p}{m} = \frac{\cdot 78}{\cdot 975} = \cdot 800$. $L^{\frac{1}{6}} = (340)^{\frac{1}{6}} = 2 \cdot 64$.

$$\begin{aligned} w &= \cdot 16[\cdot 800 \times 2 \cdot 64 - (0 \cdot 6)] \\ &= \cdot 16 \times (2 \cdot 11 - \cdot 6) \\ &= \cdot 16 \times 1 \cdot 51 = 242. \end{aligned}$$

(4) T.S.S. ——. $418 \cdot 5 \times 52 \cdot 2 \times 23 \cdot 5$ ft. mean draught. $\Delta = 9\,300$ Block coefficient $= \cdot 634$. Mid-area coefficient $= \cdot 956$. Prismatic coefficient $= \cdot 664$. $\frac{p}{m} = \cdot 964$. $L^{\frac{1}{6}} = 2 \cdot 733$.

$$\begin{aligned} w &= \cdot 13(\cdot 694 \times 2 \cdot 733 - 1 \cdot 1) \\ &= \cdot 13(1 \cdot 898 - 1 \cdot 1) = \cdot 13 \times \cdot 798 \\ &= \cdot 103\,7. \end{aligned}$$

(5) S.S. ——. $322 \times 42 \cdot 3 \times 22 \cdot 33$ ft. mean draught. $\Delta = 6\,730$. Block coefficient $= \cdot 778$. Mid-area coefficient $= \cdot 983$. Prismatic coefficient $= \cdot 791$. $\frac{p}{m} = \frac{\cdot 791}{\cdot 983} = \cdot 805$. $L^{\frac{1}{6}} = 2 \cdot 62$.

$$\begin{aligned} w &= \cdot 16 \times [\cdot 805 \times 2 \cdot 62 - \cdot 6] \\ &= \cdot 16 \times (2 \cdot 11 - \cdot 6) = \cdot 16 \times 1 \cdot 51 = \cdot 242. \end{aligned}$$

(6) S.S. ——. $355 \times 48 \cdot 7 \times 23 \cdot 5$ ft. mean draught. $\Delta = 8\,930$. Block coefficient $= \cdot 767$. Mid-area coefficient $= \cdot 976$. Prismatic coefficient $= \cdot 785$. $\frac{p}{m} = \cdot 805$. $L^{\frac{1}{6}} = 2 \cdot 66$.

$$\begin{aligned} w &= \cdot 16 \times (\cdot 805 \times 2 \cdot 66 - \cdot 6) \\ &= \cdot 16 \times 1 \cdot 51 \\ &= \cdot 242. \end{aligned}$$

(7) T.S.S. ——. $440 \cdot 3 \times 54 \cdot 1 \times 23 \cdot 5$ ft. mean draught. $\Delta = 10\,195$. Block coefficient $= \cdot 637$. Mid-area coefficient $= \cdot 973$. Prismatic coefficient $= \cdot 656$. $\frac{p}{m} = \cdot 675$. $L^{\frac{1}{6}} = 2 \cdot 757$.

$$\begin{aligned} w &= 0 \cdot 13(\cdot 675 \times 2 \cdot 757 - 1 \cdot 1) \\ &= \cdot 13(1 \cdot 86 - 1 \cdot 1) = \cdot 13 \times \cdot 76 \\ &= \cdot 098\,8. \end{aligned}$$

v = Block coefficient.	p = Prismatic coefficient.	m = Mid-section coefficient.	$\frac{p}{m}$.	Wake fraction			Twin screw or single screw.
				From Taylor's formula.	From Gordon's slide rule.	From MacDermott's formula.	
·72	...	...	...	·31	·277	...	S.S
·78	...	...	...	·34	·324	...	S.S.
·76	·78	·975	·800	·33	·31	·242	S.S.
·636	·659	·966	·681	·268	·217	·202	S.S.
·767	·785	·976	·804	·333	...	·242	S.S.
·637	·664	·956	·694	·15	·084 2	·103 7	T.S.S.
·68	·708	·961	·736	·29	...	·224	S.S.
·778	·791	·983	·805	·338	...	·242	S.S.
...	...	...	...	...	...	...	...
·637	·656	·973	·675	·15	...	·098 8	T.S.S.

In papers to the Institution of Naval Architects in 1910 and 1914, Mr W. J. Luke gave the results from experiments with models 204 in. long × 30 in. beam × 9 in. draught,—one model ·65 block coefficient, and the other ·60 block coefficient. The displacements in fresh water respectively were 1 296 lbs. and 1 175 lbs. The propeller was 6 in. diameter, having three blades, 1·2 pitch ratio, and ·375 disc area ratio.

With single screws, increasing the diameter caused a decrease in wake and an increase in thrust deduction. The hull efficiencies with the larger screws were consequently less than with the smaller screws. The performance of the screw was noted when revolving behind the full model when advancing at a speed of 332 ft. per minute (corresponding to 16 knots for a 400-ft. ship), and when "open," or apart from model, at a speed of 280 ft. per minute (estimated to be a suitable speed, allowing for wake).

Twin screws, outward turning.—With the shaft centres in standard position, the larger the screws were the greater became the wake and hull efficiencies. When revolving behind horizontal bossings the wake fraction was as high as ·32, and the hull efficiency 1·10. The resistance of the bossing was least when the web was normal to the line of shell-plating.

Full model :—

	Wake.	Thrust deduction.	Hull efficiency.
Twin screws . . .	·20	·15	1·02
Single screws . . .	·34	·17	1·11

Fine model :—

	Wake.	Thrust deduction.	Hull efficiency.
Single screws . . .	·22	·16	1·02
Twin screws . . .	·13	·13	·98

Mr Luke found that the high hull efficiencies with the twin screws in the experiments were probably due to the close proximity of relatively small propellers to the hull of a model having great beam.

TABLE XXVIII.—FOR USE WITH M'DERMOTT'S FORMULA FOR WAKE.

ω = Block coef.	p = Prismatic coef.	m = Midsection coef.	$\frac{p}{m}$.	ω = Block coef.	p = Prismatic coef.	m = Midsection coef.	$\frac{p}{m}$.
·84	·853	·986	·866	·61	·643	·950	·677
·83	·842	·985	·855	·60	·634	·947	·669
·82	·833	·985	·846	·59	·625	·944	·662
·81	·824	·984	·838	·58	·616	·942	·655
·80	·815	·983	·83	·57	·607	·940	646
·79	·805	·982	·82	·56	·598	·938	·638
·78	·795	·982	·81	·55	·588	·936	·628
·77	·786	·980	·803	·54	·58	·932	·623
·76	·775	·980	·791	·53	·57	·930	·614
·75	·768	·978	·786	·52	·561	·927	·605
·74	·758	·978	·776	·51	·554	·921	·601
·73	·749	·976	·768	·50	·548	·914	·600
·72	·739	·975	·758	·49	·542	·905	·599
·71	·73	·974	·75	·48	·54	·89	·607
·70	·721	·971	·744	·47	·539	·874	·616
·69	·711	·970	·734	·46	·537	·858	·627
·68	·703	·969	·726	·45	·54	·834	·648
·67	·694	·966	·719	·44	·54	·816	·662
·66	·686	·962	·714	·43	·543	·793	·685
·65	·678	·961	·705	·42	·55	·764	·721
·64	·67	·956	·700	·41	·565	·726	·779
·63	·66	·955	·691	·40	·587	·682	·860
·62	·651	·953	·684				

TABLE XXIX.—WAKE FRACTION FOR CALCULATIONS.
Wake fraction from curves. (Plate 66.)

Block coefficient.	Single screw.	Twin screws.		Block coefficient.	Single scew.	Twin screws.	
		a.	*b.*			*a.*	*b.*
·38	·14	·009	·01	·62	·26	·14	·117
·39	·145	·015	·015	·63	·265	·146	·121
·40	·15	·02	·02	·64	·27	·151	·125
·41	·155	·026	·024	·65	·275	·157	·13
·42	·16	·031	·029	·66	·28	·163	·134 5
·43	·165	·036	·032 5	·67	·285	·168	·138 5
·44	·17	·041 5	·037	·68	·29	·174	·143
·45	·175	·046 5	·041	·69	·295	·179	·148
·46	·18	·053	·045 5	·70	·30	·185	·152
·47	·185	·059	·05	·71	·305	·19	·156 5
48	·19	·064	·055	·72	·31	·196	·161
·49	·195	·07	·059	·73	·315	·201	·165
·50	·20	·075	·064	·74	·32	·206 5	·17
·51	·205	·08	·068	·75	·325	·213	·174
·52	·21	·086	·072	·76	·33	·218	·179
·53	·215	·091	·077	·77	·335	·224	·183
·54	·22	·097	·081	·78	·34	·229	·187
·55	·225	102	·085	·79	345	·235	·191
·56	·23	·108	·09	·80	·35	·24	·196
57	·235	·113	·094 5	·81	·355	·245	·201
·58	·24	·119	·099	·82	·36	·251	·205
·59	·245	·125	·103	·83	·365	·256	·21
·60	·25	·13	·108	·84	·37	·262	·214
·61	·255	·135	·112	·85	·375	·268	·219

Twin screws, inward turning.—Increasing the diameter only modified the wake value slightly, but steadily increased the thrust deduction and gave a much lower hull efficiency. When revolving behind horizontal bossings, the standard screws showed a wake fraction of only ·10, the hull efficiency being only ·94. Inward-turning screws with horizontal bossings gave poor results.

The experiments with twin screws generally showed, when trying different transverse positions, that the closer they "were placed to the hull the higher became the wake and hull-efficiency values. Experiments dealing with fore-and-aft position indicated that the further aft the screws were placed the less became the wake and thrust-deduction values.

"Decrease in the value of the hull-efficiency elements accom-

panied decrease in pitch ratio, and neither area nor number of blades had any appreciable effect on wake and thrust deduction."

Angle of bossing has a considerable influence upon the effect of the wake. Mr Luke's 1910 paper to the I.N.A. mentions that "with horizontal bossings and propellers turning outwards, a large wake results, which decreases steadily with increase of slope of web. With propellers inward turning just the opposite effect is apparent."

Various angles of shaft bossing show almost equal thrust-deduction values. Consequently, the hull-efficiency values show considerable variation with different bossing angles. When the model is towed at standard speed without the propellers, greatest resistance accompanies horizontal bossings, and an angle of 45° from the horizonal offers minimum resistance. Mr Luke found that "an outward-turning screw should have bossings of less angle than the slope associated with least resistance, and if an inward-turning screw be used, a steep angle of bossing should be adopted, in order to avoid a low hull-efficiency value."

Mr Luke found that when the pitch of the propellers was increased, the wake and thrust deduction were both slightly increased, the resulting hull efficiency remaining practically constant; and that when twin screws were given different clearances from the hull—whether brought about by spreading the shafts farther apart or by varying the fore-and-aft position of the screws,—wide clearances gave diminished wake gain, "but as an offset produced less thrust deduction than those obtained when the propellers were brought close to the hull." Well-arranged bossings might actually give substantially greater hull-efficiency value than would be obtained with no bossings, but any such gain was neutralised, if not exceeded, by the increase in hull resistance due to these appendages.

The wake value is affected by the size of the screw to some extent, because the speed of the wake varies roughly in the manner shown on Plate 66, as the distance from the centre of the ship is increased, so that a small single screw works in a greater wake than a larger screw on the same ship. Mr Baker has pointed out that this result does not apply to a ship with a very full stern, owing to the dead-water effect. The smaller screw would have a larger slip, and probably lower screw efficiency, which would tend to discount the apparent gain in hull efficiency. The wake and hull efficiency have been found by Mr Luke to decrease very slightly for a given ship as the speed is increased, but the variation is unimportant.

DUTCH OPIUM CRUISER "ARGUS."

(Particulars from Dr Robert Caird's Trial Analysis Curves.)

Knots.	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}}$.	Percentage of designed full speed of 16 knots.	Lbs. Mean pressure referred to L.P. cylinder.	Apparent slip per cent.	Wake factor wV, where V = ship speed.	(e_1) $\frac{\text{D.H.P.}}{\text{I.H.P.}}$.	(e_2) Propeller efficiency.	Propulsive coefficient.	Percentage of full-power revolutions for 16 knots.	Real slip per cent. of ship speed.	(e_3) Hull efficiency.
6	187	37·5	5·8	15·3	·251	·535	·628	·452	34·4	36·9	1·34
8	241	50	8·12	13·4	·202	·63	·666	·524	46·3	31	1·26
10	267	62·5	11·3	12·0	·19	·708	·675	·569	58·5	28·8	1·194
12	269	75	13·7	11·3	·20	·77	·673	·593	71·3	28·8	1·145
14	253	87·5	22·9	12·0	·218	·82	·663	·60	84·3	31·1	1·11
16	220	100	33·3	14·1	·241	·853	·645	·594	100	34·9	1·090
17	199	106·3	41	...	...	·865	...	·59	108·2	...	1·085

Symbols and Working Formulæ for Propellers.

Let T = the thrust of the screw in lbs.
T.H.P. = thrust horse-power of the screw.
V_A = the speed of advance of the propeller in knots through the wake water in which it works.

Then

$$T = \frac{\text{T.H.P.} \times 33\,000}{\text{Speed of advance of propeller in feet per min.}},$$

$$T = \frac{\text{T.H.P.} \times 33\,000}{V_A \times 101 \cdot 33},$$

or

$$T = \frac{\text{T.H.P.} \times 60 \times 33\,000}{V_A \times 6\,080},$$

$$T = \frac{\text{T.H.P.} \times 325 \cdot 66}{V_A},$$

$$T = \frac{\text{T.H.P.}}{V_A \times \cdot 003\,070\,7}.$$

$$V_A = V_S - wV_S,$$

where V_S = speed of ship in knots.
w = wake fraction.

(1) Real slip ratio = S_2

$$= \frac{\text{Speed of propeller in feet per min.} - \text{speed of advance of propeller in feet per min.}}{\text{Speed of propeller in feet per min.}}$$

$$= \frac{PR - V_A}{PR}.$$

V^A = speed of advance of propeller through the wake water in which it works.
V_S = speed of ship.
w = wake fraction.
$V_A = V_S - wV_S$.

Taylor's formulæ for wake fraction :—

For single screws, $w = -\cdot 05 - \cdot 5b$, where b = block coefficient.
For twin screws, $w = -\cdot 2 + \cdot 55b$.

Example.—If $w = \cdot 333$, $V_A = 6 \cdot 835$ knots where $V_S = 10 \cdot 25$ knots, or $V_A = 6 \cdot 835 \times 101 \cdot 33 = 692$ ft. per min.

If revs. per min. = $R = 66$, $PR = 1\,081$ where $P = 16 \cdot 4$ ft. pitch.

Then S_2 = ·358 or 35·8 per cent.

(2) Another formula for S_2:

$$S_2 = S_1 + \frac{v_0}{PR}$$

where v_0 = wake speed in feet per min.

S_1 = apparent slip.

S_1 = Apparent slip ratio = $\frac{\text{Speed of propeller} - \text{speed of ship}}{\text{Speed of propeller}}$ all in feet per min.

$$= \frac{PR - (V_S \times 101{\cdot}33)}{PR} = \frac{1\,081 - (10{\cdot}25 \times 101{\cdot}33)}{1\,081} = {\cdot}038\,8.$$

or 3·88 per cent. $v_0 = w \times (V_S \times 101{\cdot}33)$.

If $v_0 = {\cdot}333 \times (10{\cdot}25 \times 101{\cdot}33) = {\cdot}333 \times 1\,039 = 346$ ft. per min., then $S_2 = S_1 + \frac{v_0}{PR} = {\cdot}038\,8 + \frac{346}{1\,081} = {\cdot}358$, or 35·8 per cent. as before.

(3) We may write

$$S_2 = S_1 + v_0.$$

Real slip in feet per min. = apparent slip in feet per min. + wake speed in feet per min.,

or

Real slip in knots = apparent slip in knots + wake speed in knots.

Three formulæ for real slip :—

Let S_1 = apparent slip ratio.

p = pitch of propeller in feet.

N = revolutions per min.

v_0 = wake speed in feet per min. = $w \times (V_S \times 101{\cdot}33)$.

S = real slip ratio.

$100 \times S_1$ = apparent slip per cent.

$100 \times S$ = real slip per cent.

w = wake fraction.

wV_S = wake speed in knots.

V_S = speed of ship in knots.

V = speed of advance of propeller (through wake) in knots.

v_S = speed of ship in feet per min. = $V_S \times 101{\cdot}33$.

v = speed of advance of propeller in feet per min. = $V \times 101{\cdot}33$.

Then for real slip ratio we have the three formulæ :—

$$\text{(I.)} \qquad S = S_1 + \frac{v_0}{pN},$$

$$\text{(II.)} \quad 1 - s = (1 - s_1)(1 - w),$$

$$\text{(III.)} \qquad s = \frac{pN - (v_s - wv_s)}{pN},$$

or

$$S = \frac{pN - (V_S - wV_S)101 \cdot 33}{pN}.$$

THRUST. (R. E. Froude's formulæ.)

$$T = D^2V^2 \times B\frac{p+21}{p} \times \frac{1 \cdot 02S(1 - \cdot 08S)}{(1-S)^2}$$

$$T = aD^4R^2S \times 1 \cdot 02(1 - \cdot 08S),$$

where D = diameter in feet.
B = blade factor (see Table XXXII).
S = real slip ratio.
V = speed of advance in feet per min.
$p = \frac{p}{D}$ = pitch ratio.

These provide the key figure for propeller analysis and design, viz. thrust in lbs. Its relation to thrust horse-power is shown on p. 161.

Referring to Taylor's elliptical blades, the relation between mean-width ratio and area ratio is roughly somewhat as follows :—

Mean-width ratio.	Number of blades.	Area ratio: With solid propeller.	Area ratio: With built propeller.
·15	4	·287	·276
·20	4	·398	·383
·20	3	·298	·288
·25	3	·374	·36
·25	4	·498	·479
·30	3	·451	·435
·30	4	·603	·58
·35	3	·54	·519

With the thumb-shaped blade (a rather wider-tipped ellipse), of mean-width ratio = ·196, the area ratio with solid propeller would be ·387, and with solid propeller about ·403. In this type the mean width would be about $3\frac{1}{2}$ per cent. greater than with the elliptical blade. A solid propeller would have 4 per cent. greater blade area than a built propeller, the boss being smaller. With a built propeller, with cast-iron boss and the blades recessed in, the radius from the shaft centre to the part of the blade at which the net surface begins would be about ·265 of the half diameter of the propeller. For a built propeller with cast-steel boss, the figure would be about ·23. For a cast-iron solid propeller it would be about $3\frac{1}{2}$ or 4 per cent less.

In analyses of progressive trials, where the propeller efficiency is found to be lower by $4\frac{1}{2}$ per cent. or so than in the tables, this may be due to blunt blade edges, or to the inclusion of air resistance in the thrust, or to the effect of a want of homogeneity in the wake. Models are tried in open water, while actual screws work in wake more or less disturbed, *i.e.* in water moving past the screw in an undefined way.

$$\text{Froude's } 1{\cdot}02 = \frac{\text{Effective pitch for naked hull E.H.P.}}{\text{Face pitch for total E.H.P., including appendages and air}}$$

The figure 1·02 seems to answer with trial trip results, *i.e.* smooth-water trials, or with E.H.P. computed from Taylor's contours, with perhaps a percentage addition to the E.H.P. to bring the results from American temperatures into line with average sea-water conditions.

When using I.H.P., the engine efficiency e_1 should be taken from Plate 40. Thus D.H.P. = I.H.P. $\times e_1$, where e_1 is the product of the brake H.P. of the engine $\times$ shaft transmission efficiency. ·82 is a value of e_1 used by Mr Denholm Young, and is an average figure for cargo reciprocating engines at sea, *i.e.* including appendages, air, and weather, at seven-eighths or nine-tenths full power, with engine-driven pumps. For these conditions 1·03, 1·05, or 1·09 may be found to give values of C_A and C_O agreeing with Froude's results.

Selection of the Principal Dimensions of a Propeller.

The usual propeller problem is to select dimensions suitable for driving the ship at a given speed with given revolutions of the main engines. There are in common use two methods of estimating the dimensions which will develop the thrust necessary to drive the ship. One is based on the water resistance of the naked hull of the ship; the other on the total resistance, in-

cluding appendage and air resistance. Either method is sound in principle, and the one which should be selected will depend upon the form in which the basic information is available. The former is a good one if the E.H.P. derived from tank experiments on the model of the hull is available, since in passing from one ship of known performance to another with similar means of propulsion it is a most reliable guide in settling the propulsive coefficient. The latter is perhaps the one in more common use where model experiment data are not to hand. It involves an estimate of the naked hull as well as the appendage, and sometimes air resistance. At the best one can only make a jump in the dark at the two latter, and the estimates of the former, according to empirical rules suggested by various people, are often very disconcerting. It has, however, the merit that some account, albeit probably inaccurate, is taken of resistances which must exist in practice.

Whatever method be selected, having settled upon the effective horse-power whether for naked hull or for hull and appendages, an estimate is made from propeller characteristic curves—thrust, slip, and efficiency—for model screws using effective pitch (*not* face pitch) of diameter pitch and area of a screw propeller which will develop the estimated horse-power whether for naked hull or for hull and appendages. Practical conditions will generally determine the diameter of the screw. The pitches corresponding to the same diameter will therefore differ by the two methods, since the horse-power to be developed will be different. In the ship the propellers must do the same work, so that in passing from the estimated effective pitch to the selected face pitch a different coefficient must be used in the two methods. Mr Froude gives in his 1908 paper a factor based on a careful analysis in which account was taken of total resistance of progressive trial results of twin-screw warships expressing the relation between *effective pitch* necessary to develop *naked hull* horse-power and the *face pitch* necessary to develop *total* E.H.P. in the ship. He states that this effective pitch is equal to 1·02 times the ship face pitch. Consequently, if the second method of calculation is used, a factor greater than 1·02 must be used, since the pitch necessary to overcome the total resistance will be greater than that corresponding to the naked-hull resistance. This factor can only be determined from an analysis of trial results, but it will more nearly agree with the actual relation of effective pitch to face pitch of individual screws, curves of which, derived by a careful analysis of Mr Taylor's experiments, were given by Mr T. B. Abell in the *Transactions of the Institution of Naval Architects*, 1908.

In other words, the analysis pitch should be taken as 1·02 times the nominal (or driving face) pitch for ship.

Whether we adhere to Mr Froude's 1·02 or not depends upon conditions of running, width of blade, and blade thickness fraction. 1·05 to 1·09 have been found to give values agreeing with Mr Froude's results. 1·09 is not intended to be a measure of the ratio of effective pitch to nominal pitch,—it is only a factor used in comparing Froude's figures with realisations in actual ships, and probably depends upon the speed of advance as much as on anything.*

The usual method of using the $C_A\, C_O$ data "is to obtain diameter and efficiency for two or more of the pitch ratios for which curves are given, each for two or more values of disc-area ratio, and plot the results on a base of total blade area. In this way the diameter and efficiency for any intermediate pitch ratio is indicated," remembering the discount to be made to allow for portion of area covered by boss.

$$C_A = \frac{R^2H}{BV^5}\left(= \frac{p+21}{p^3} \,.\, x^2y\right).$$

$$C_O = \frac{H}{BD^2V^3}\left(= \frac{p+21}{p} \,.\, y\right).$$

$$p = \text{pitch ratio} = \frac{P}{D}.$$

$$r = \frac{\text{revolutions}}{100}.$$

$$H = \text{thrust H.P.}$$

$$V = \text{speed of advance.}$$

In an interesting address to the Institute of Marine Engineers on 7th September 1915, Sir Archibald Denny, Bart., gave an account of experiments to ascertain the discrepancy between the real pitch and that of the driving face, showing that it varied with the speed of advance as well as with the width and shape of the blade, and with its thickness. Experiments to find the effect of revolutions alone showed that real pitch did not remain the same throughout all revolutions and thrusts in the actual propeller.

Professor T. B. Abell showed in 1910 (*Trans. I.N.A.*) how the effective pitch differed for different speeds, and gave curves, plotted to a base of disc-area ratio, to show the resulting ratio of effective to face pitch for the different three-bladed screws of Mr Taylor's

* Perhaps when the influence of speed of advance upon ratio of effective pitch to nominal pitch has been further investigated by experiment, another method will be found which will give a more satisfactory general solution.

TABLE XXX.

Slip ratio.	x.	y.	Slip ratio.	x.	y.
0	1·013	0	·26	1·370	·001 495
·02	1·034	·000 067	·28	1·407	·001 698
·04	1·056	·000 139	·30	1·448	·001 922
·06	1·078	·000 217	·32	1·491	·002 169
·08	1·101	·000 302	·34	1·535	·002 442
·10	1·126	·000 394	·36	1·583	·002 745
·12	1·152	·000 494	·38	1·635	·003 086
·14	1·178	·000 602	·40	1·689	·003 457
·16	1·206	·000 720	·42	1·747	·003 880
·18	1·236	·000 849	·44	1·810	·004 345
·20	1·267	·000 989	·46	1·877	·004 887
·22	1·299	·001 142	·48	1·949	·005 49
·24	1·333	·001 311	·50	2·027	·006 175

experiments. In these propellers the boss diameter was ·195 5 of the propeller diameter, and the $\dfrac{\text{Root thickness}}{\text{Length of blade to root}} = \dfrac{1}{23}$.

For these curves the ordinates $\dfrac{\text{Effective pitch}}{\text{Face pitch}}$ ran as in the following table:—

TABLE XXXI.

Disc-area ratio.	$\frac{\text{Effective pitch}}{\text{Face pitch}}$ for various pitch ratios ·6 to 1·4, thus:				
	·6.	·8.	1·0.	1·2.	1·4.
·20	1·337 2	1·203 9	1·141 6	1·118 8	1·083 3
·25					
·30	1·245	1·155	1·122 5	1·105	1·074
·35	1·208	1·136			
·40	1·18	1·122 5	1·102	1·086	1·065
·45	1·155	1·11	1·093	1·08	1·062
·50	1·133	1·10	1·083	1 071	1·057 5
·55	1·112	1·088	1·073		
·60	1·105	1·075	1·065	1·056 5	1·052 5
·65	1·091	1·062	1·056 5	1·052	1·049

Figs. 44 and 68 show other estimates.

Mr A. M. Gordon's propeller slide rule, the accuracy of which has been verified at Haslar, gives suitable propeller dimensions based upon Froude's 1908 paper, which marked a distinct step in advance of the methods of the 1886 and 1892 papers.

In using Mr A. M. Gordon's propeller slide rule, the D.H.P. should be used instead of the I.H.P. For a single-screw cargo steamer, take e_1 = ·84, making D.H.P. = ·84 × I.H.P. for sizes about 2 000 horse-power. The speed of the ship may be taken as the speed at load draught at sea in moderate weather.

For a passenger liner, the same conditions; but if the vessel is of finer block than usual for the speed, the propeller may come out a little larger than ordinary practice, on account of the lower wake fraction attending the finer form, and the desired result is most likely to be obtained if a lower propulsive coefficient than ·50 is taken.

Examples.—Single-screw cargo vessel, 10¼ knots at 72 revolutions, corrected speed = 7·11, 1 900 I.H.P. Pitch ratio = ·956. Propeller diameter = 16 ft. 9 in. Four blades. Pitch = 16 ft. 0 in. Area ratio = ·40. D.H.P. = 1 600. Efficiency about 64 per cent. Ship 340 × 46½ × 23⅓ ft. draught. Block coefficient = ·76. Propulsive coefficient = ·50.

Astern Power.

On trial, 85 revolutions per minute; 2 500 I.H.P. when going ahead. When put astern with engine stop valve the same amount open, the revolutions per minute went up to 104; 3 000 I.H.P. Astern T.H.P. probably roughly ·89 ahead.

Twin-screw steamer, 418 × 52 × 23 ft. draught, 14½ knots, 4 650 I.H.P. total. 75 revolutions. Propulsive coefficient = ·45. Block coefficient = ·637. Efficiency = 71·2 per cent. Corrected speed = 13·22. Diameter = 16 ft. 9 in. Three blades. Pitch = 21 ft. Area ratio = ·352. For area ratio = ·375. Diameter = 16 ft. 8 in. Pitch = 20 ft. 11 in.

Single-screw cargo vessel, 375 × 51·7 × 25 ft. mean draught. Block coefficient = ·76. 11 knots at 72 revolutions. 2 350 I.H.P. D.H.P. = 1 910. Corrected speed = 7·6. Pitch ratio = ·945. Efficiency = 6 485 per cent. Diameter = 17 ft. 6 in. Pitch = 16 ft. 6 in. Area ratio = ·40. Four blades. For area ratio = ·375, diameter = 17 ft. 9 in., pitch = 16 ft. ½ in. for same efficiency.

S.S. ——. Four-bladed cast-iron solid propeller. Blade thickness fraction = ·515. Expanded area ratio = ·40. Two published diagrams give ratios of effective pitch to nominal face pitch,

viz. Professor T. B. Abell's fig. 4 (Institution of Naval Architects, 1910), and Mr T. C. Tobin's fig. 2 (Institution of Naval Architects, 1916). Both deal with three-bladed propellers. For this S.S., with four blades, the boss would be the same size relatively as in Mr Taylor's propellers, to which both the above diagrams refer. The disc-area ratio, however, would first have to be multiplied by $\frac{3}{4}$ to use the three-blade diagram for a four-bladed screw, as the pitch correction is based upon width of blade corresponding to area ratio. A four-bladed propeller of ·40 area ratio would have blades of the same width as a three-bladed propeller of ·30 area ratio, and the same pitch correction, or ratio of effective pitch to face pitch. The projected area would be $\cdot 84 \times \cdot 30 = \cdot 252$.

For blade thickness fraction ·515, Mr Tobin's fig. 2 gives ratio of effective pitch to face pitch $= 1{\cdot}186$.

Professor T. B. Abell's fig. 4 gives effective pitch ÷ face pitch $= 1{\cdot}121$.

Our Plate 44, however, gives ratios of effective pitch ÷ face pitch lower than the above, viz. 1·02 to 1·1 say.

In Froude's models $\frac{\text{Root thickness}}{\text{Blade length}} = \frac{1}{16{\cdot}5}$, which means a very thin blade. In ordinary merchant-ship propellers with cast-steel or bronze blades $\frac{1}{8}$ is usual. With cast-iron solid propellers the thickness of course is greater, and for these a higher multiplier than 1·02 should be used in connection with Froude's method of applying model data to full-sized propellers.

T.S.S. ——. 7 760 I.H.P. at 14·52 knots. 96 revolutions. Apparent slip per cent. $= 9{\cdot}83$. $\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = 288$. Block coefficient $= \cdot 724$. Propellers built, three-bladed. D. = 17. P. = 17. Expanded area ratio $= \cdot 415\,5$. $w = -\cdot 2 + (\cdot 55 b) = -\cdot 2 + (\cdot 55 \times \cdot 724) = -\cdot 2 + \cdot 398 = \cdot 198$. $(V_S \times 101{\cdot}33) = 14{\cdot}52 \times 101{\cdot}33 = 1\,472$. Take effective pitch ÷ face pitch $= 1{\cdot}02$. Effective pitch ratio $= 1{\cdot}02$. Take "B" $= \cdot 109\,0$. $p =$ effective pitch $= 17{\cdot}35$. $N = 96$. $pN = 17{\cdot}35 \times 96 = 1\,668$.

$$S_1 = \frac{pN - 1\,472}{pN} = \frac{1\,668 - 1\,472}{1\,668} = \cdot 117\,7.$$

$V = V_S - wV_S = 14{\cdot}52 - (\cdot 198 \times 14{\cdot}52) = 11{\cdot}64$ knots speed of advance.

$v_0 = w \times (V_S \times 101{\cdot}33) = \cdot 198 \times 1\,472 = 292.$

$$\text{Real slip} = S_2 = S_1 + \frac{v_0}{pN} = \cdot 117\,7 + \frac{292}{1\,668} = \cdot 292\,8.$$

Efficiency (Froude, 1908) $= \cdot 691 + \cdot 002\,5 = \cdot 693\,5$.

$$\text{D.H.P.} = \cdot 865 \times \frac{7\,750}{2} = 3\,350. \quad H = \cdot 693\,5 \times 3\,350 = 2\,320.$$

$$V^5 = 213\,700. \quad V^3 = 1\,577. \quad D^2 = (17)^2 = 289.$$

$$C_A = \frac{(\cdot 96)^2 \times 2\,320}{\cdot 109\,0 \times 213\,700} = \cdot 092,$$

or with $B = \cdot 112\,3$, $C_A = \cdot 089\,3$.

$$C_O = \frac{2\,320}{\cdot 109\,0 \times 289 \times 1\,577} = \cdot 046\,6,$$

or with $B = \cdot 112\,3$, $C_O = \cdot 045\,3$.

These values are about 16 per cent. too high, which may be because we have taken too low a ratio of effective pitch to nominal pitch, or because the ratio of "B" values for 4 blades to "B" values for 3 blades is not certain.

Try effective pitch÷nominal pitch = 1·06. Effective pitch ratio = 1·06. Effective pitch = 18·01. If blades are bluff-tipped ellipses, B = ·112 3. $pN = 18{\cdot}01 \times 96 = 173\,0$.

$$S_1 = \frac{1\,730 - 147\,2}{1\,730} = \cdot 149.$$

$$S_2 = S_1 + \frac{v_0}{pN} = \cdot 149 + \frac{292}{1\,730} = \cdot 317\,9.$$

Efficiency (Froude, 1908) = ·682 + ·002 5 = ·684 5.

$$H = \cdot 684\,5 \times 3\,350 = 2\,292.$$

$$C_A = \frac{(\cdot 96)^2 \times 2\,292}{\cdot 112\,3 \times 213\,700} = \cdot 088\,2.$$

$$C_O = \frac{2\,292}{\cdot 112\,3 \times 289 \times 1\,577} = \cdot 044\,8.$$

Not more than ½ per cent. in error.

S.S. "Anselm." 400 × 50 × 23 ft. mean draught. Single screw. 3 840 I.H.P., 75 revolutions, 14 knots. Four blades bronze, steel boss. D. = 19. Face pitch = 20·5. Expanded area ratio = ·352. Projected area ratio = ·295 5. Face-pitch ratio = 1·08. Face-pitch apparent slip = 7·8 per cent. (14 × 101·33) = 1 419. Block coefficient = ·68. w = ·29. For modified Tobin's diagram for three blades, we may consider the blade areas and widths = ¾ths those of a three-bladed propeller of same area ratio, viz. expanded area ratio = ·264, and projected area ratio = ·221 5. As it is a built propeller, the larger boss will make the blade areas and widths 6½ per cent. greater than in Taylor's propeller, where the boss is as in a solid propeller, viz. expanded area ratio = ·283 5, and projected area ratio = ·236. For these we have effective pitch÷face pitch = 1·07. Effective pitch ratio = 1·156. Effective pitch = 21·96. pN = 1 645.

$S_1 = \cdot 187\,5$. $v_0 = \cdot 29 \times 1\,419 = 411$. $V = 9 \cdot 94$ speed of advance.
$S_2 = \cdot 387\,5$.

The C_A and C_O values so obtained do not agree with Froude's data. Let us lay aside effective pitch corrections and try Mr Froude's 1·02 as a multiplier for pitch. Pitch ratio = 1·101.

$$p = 20 \cdot 5 \times 1 \cdot 101 = 20 \cdot 92.$$
$$S_1 = \cdot 096\,2.$$
$$S_2 = \cdot 096\,2 + \frac{411}{1\,570} = \cdot 358\,2.$$

Efficiency = $\cdot 66 + \cdot 004 - \cdot 012\,5 = \cdot 651\,5$. D.H.P. = $\cdot 84 \times 3\,840 = 3\,225$. $H = \cdot 651\,5 \times 3\,225 = 2\,100$.

$$\left.\begin{aligned} C_A &= \frac{(\cdot 75)^2 \times 2\,100}{\cdot 109\,0 \times 96\,900} = \cdot 112. \\ C_O &= \frac{2\,100}{\cdot 109\,0 \times 361 \times 982} = \cdot 054\,2. \end{aligned}\right\} \text{Agreeing with Mr Froude's values.}$$

The "B" value for an elliptical blade, with 20 per cent. allowance for boss, agrees with the data.

Calculation of Propeller Dimensions.

By Mr R. E. Froude's $C_A\,C_O$ Constants (*Trans. Inst. N.A.*, 1908).

Example 1.—S.S. "Justin," single-screw steamer, 355 × 48·7 × 23·5 ft. mean draught. Block coefficient = ·767. 1 850 I.H.P., 10½ knots, 66 revolutions. Propeller, four-bladed cast-iron solid. Diameter = 17 ft. Pitch = 17 ft. Pitch ratio = 1·0. Expanded area ratio = ·40. Blade thickness fraction = ·515. Let diameter = D. Pitch = p. Revolutions per min. = N. V_S = speed of ship in knots. V = speed of advance of propeller. Wake fraction by Taylor's formula $w = -\cdot 05 + (\cdot 5 \times b) = \cdot 333$. b = block coefficient. Using Professor T. B. Abell's 1910, fig. 4, ratio of effective pitch to face pitch = 1·09. Effective pitch = 18·52 ft. Effective pitch ratio = 1·09. $pN = 1\,222$. Apparent slip = $S_1 = \dfrac{pN - (V_S \times 101 \cdot 33)}{pN} = \dfrac{1\,222 - 1\,065}{1\,222} = 1\,285.$

$V = V_S - wV_S = 10 \cdot 5 - (\cdot 333 \times 10 \cdot 5) = 7$ knots, speed of advance.
Real slip = S_2.

$v_0 = w \times (V_S \times 101 \cdot 33) = \cdot 333 \times 1\,065 = 355$ = wake speed in ft. per min.

$$S_2 = S_1 + \frac{v_0}{pN} = \cdot 128\,5 + \frac{355}{1\,222} = \cdot 418\,5.$$

Efficiency.—Froude's 1908 tables give ·611 for three-bladed propeller with ·45 area ratio, taking whole ellipse. The boss brings our area ratio ·40 equivalent to about the same figure, viz. ·454. Deduct ·012 5 to correct the efficiency for four blades, making efficiency $(e_2) = \cdot 598\,5$. Now take D.H.P. = ·835 × I.H.P. = 1 546. Thrust H.P. = H = ·598 5 × 1 546 = 924.

$$C_A = \frac{(\cdot 66)^2 \times 924}{\cdot 120\,3 \times 16\,800} = \cdot 199.$$

$$C_O = \frac{924}{\cdot 120\,3 \times D^2 \times 343} = \cdot 077\,4.$$

Both of these values agree with Mr Froude's tables. The "B" values are corrected for boss allowance by our Plate 52.

Or

$$C_A = \frac{(\cdot 66)^2 \times 953}{\cdot 123\,8 \times 17\,200} = \cdot 195\,1. \quad \text{1 per cent. too low.}$$

$$C_O = \frac{953}{\cdot 123\,8 \times 289 \times 348} = \cdot 076\,4. \quad 1\tfrac{1}{2} \text{ per cent. too low.}$$

Single-screw steamer, 322 × 42·3 ft. beam × 22·33 ft. mean draught. $\Delta = 6\,740$. Block coefficient = ·776. $w = \cdot 338$. Four-bladed cast-iron solid propeller. D. = 15·5. Nominal pitch = 16·5. Area ratio = ·383. Face pitch = 1·065. $9\tfrac{5}{8}$ knots, 1 300 I.H.P., 68 revolutions. Try effective pitch ÷ face pitch = 1·02. Effective pitch ratio = 1·065 × 1·02 = 1·087. B.T.F. = ·056 5. Effective pitch = 16·83 ft. $pN = 16{\cdot}83 \times 68 = 1\,145$.

$$S_1 = \frac{1\,145 - 975}{1\,145} = \cdot 148\,6.$$

$$S_2 = \cdot 148\,6 + \frac{329{\cdot}5}{1\,145} = \cdot 436\,6.$$

$$\begin{aligned} v_0 &= w \times (V_S \times 101{\cdot}33) \\ &= \cdot 338 \times (9{\cdot}625 \times 101{\cdot}33) \\ &= 329{\cdot}5. \end{aligned}$$

Efficiency (Froude, 1908) = ·595 + ·003 − ·012 5 = ·585 5.

$$H = 1\,071 \times \cdot 585\,5 = \cdot 627. \qquad \text{D.H.P.} = \cdot 825 \times 1\,300 = 1\,071.$$

$$C_A = \frac{(\cdot 68^2) \times 672}{\cdot 124\,3 \times 10\,540} = \cdot 237. \quad C_O = \frac{672}{\cdot 124\,3 \times 240 \times 259} = \cdot 086\,9.$$

These values of C_A and C_O agree exactly with Mr R. E. Froude's tables.

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = 246.$$

Determination of Screw-Propeller Dimensions.

The leading methods of investigating propeller dimensions are based upon facts observed in experiments with actual propellers and model propellers. The experiments enable us to determine the thrust or push forward of a propeller of a given type at any speed of ship, pitch ratio, diameter, and revolutions per minute. The thrust values from Mr R. E. Froude's experimental data are represented by his formula

$$T = aD^4R^2S \times 1{\cdot}02(1 - {\cdot}08s),$$

where T = thrust in lbs.

a is proportional to $p(p+21)$, where p = pitch ratio (see Plate 67).
D = diameter in feet.
R = revolutions per minute.
S = real slip ratio.

The two main classes of methods differ in the manner in which the thrust or thrust horse-power is estimated :—

(1) Where the T.H.P. delivered by the propeller, which is usually slightly in excess of the E.H.P., is estimated from the E.H.P. by applying wake and thrust-deduction factors obtained from analyses of progressive trials ; and

(2) in which the T.H.P. is obtained by multiplying the S.H.P. or D.H.P. by the propeller efficiency. T.H.P. = $e_2 \times$ D.H.P., where e_2 = propeller efficiency from Plates 50–51, based upon real slip ratio.

The choice of a method of designing the propeller depends upon the way we get the figure for power. In the first method, the E.H.P. is supposed to be obtained from (*a*) a tank trial ; or, failing that (*b*), by calculation, using Taylor's contours for residuary resistance, adding 5 per cent. perhaps, and using our Table X. for skin H.P., and giving an overall percentage addition to provide for appendages; or (*c*), from C values from tank trials of ships as nearly similar as possible to our own.

The first method (*b*), however, is one which enables us to calculate the E.H.P. (naked), which may be employed as the numerator in the "nominal efficiency of propulsion," where the denominator is the I.H.P. or S.H.P. from actual service running. The E.H.P. (naked) = skin H.P. calculated from our tables (Tideman's constants) + residuary H.P. from Taylor's contours.

In the second method, the gross I.H.P. or S.H.P., or B.H.P. or

D.H.P., is estimated by the Admiralty coefficient. This is the favourite rough-and-ready method of estimating power. A skilled and practised estimator may handle the Admiralty formula with such precision that it is at least allowed to be the final check in most offices.

In both methods T.H.P. is the propeller power. In the first method, T.H.P. = E.H.P. × a multiplier representing wake gain and thrust deduction. In the second method, T.H.P. = D.H.P. × propeller efficiency.

In the first method (*a*), $\frac{\text{E.H.P. (naked)}}{\text{I.H.P.}}$ = propulsive coefficient.

In the first method (*c*), $\frac{\text{E.H.P. (naked)}}{\text{I.H.P. or S.H.P.}}$ = propulsive coefficient.

In the first method (*b*), $\frac{\text{E.H.P. (naked) calculated}}{\text{I.H.P. or S.H.P.}}$ = "nominal efficiency of propulsion," or calculated propulsive coefficient.

To calculate the gross T.H.P., which is the figure we require for propeller calculations, take the following example :—

Let E.H.P. (naked) = 2 300 (made up of skin H.P. = 1 800 and residuary H.P. = 500). Air H.P. = 300. Hull efficiency = ·99. Appendage allowance = 9 per cent. of E.H.P. (naked). Then the T.H.P. naked and under tank conditions = $\frac{\text{E.H.P. (naked)}}{\text{Hull efficiency}}$ = $\frac{2\,300}{\cdot 99}$, and the gross T.H.P. = $\frac{2\,300}{\cdot 99} + 300 + (9 \text{ per cent.} \times 2\,300)$.

We may calculate the "nominal propulsive coefficient," for a series of vessels of which we know the dimensions and performances on trial or on service, and apply this "nominal propulsive efficiency" to calculate the I.H.P. or S.H.P. for a proposed ship. If we omit the appendage resistance and air resistance from the numerators of the type ships, we omit these additions from the corresponding figure for the proposed ship.

e_2, the propeller efficiency, is plotted upon a base of S, the real slip ratio, which is usually calculated from figures for the wake and ship's speed (see p. 162).

For unity pitch ratio, "B" of Froude = "*a*" ÷ 22.

In all cases

$$\text{“}a\text{”} \propto p(p+21) \quad . \quad . \quad . \quad . \quad (1)$$

$$\text{“}a\text{”} = \text{B} \times p(p+21) \quad . \quad . \quad . \quad . \quad (2)$$

$$p(p+21) = \frac{\text{“}a\text{”}}{\text{B}} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$\text{B} = \frac{\text{“}a\text{”}}{p(p+21)} \quad . \quad . \quad . \quad . \quad (4)$$

The ratio of the diameter of the boss to the diameter of the propeller should be taken into account when selecting constants which depend upon area ratio. The boss of a solid propeller cuts off roughly about 13½ per cent. of the area of the complete ellipse of blade contour, while the boss of a built propeller cuts off somewhere about 20 per cent. of the total area of the ellipse whose major axis equals the radius of the propeller.

Froude's "B" values and curves for efficiency correction are based upon area ratios which refer to the area of the whole ellipse. As there is a mean-width ratio corresponding to each area ratio, it is mean-width ratio which we ought to keep in mind when using constants to suit different blade-area ratios. For a standard form of blade outline, as the breadth of the blade at the root fillet bears a fixed relation to the width ratio and the area ratio, the ratio of effective pitch to nominal face pitch requires the same adjustment to the boss diameter as the "B" values, efficiency corrections, and other constants which depend upon expanded area ratio. Some curves showing the ratio of effective pitch to nominal face pitch for different area ratios and pitch ratios for Taylor's experimented three-bladed screws were given by T. B. Abell at the Institution of Naval Architects, 1910, and another set of curves for converting nominal pitch to effective pitch for Taylor's three-bladed model screws appeared in Mr T. C. Tobin's paper to the I.N.A., 1916, entitled "Note on Maximum Propulsive Efficiency of Screw Propellers." In both of these publications the area ratio was that of an actual screw having boss diameter = ·2 × propeller diameter, as in Taylor's experiments.

In using Froude's "B" values and efficiency correction, the figures for expanded area ratio and mean-width ratio of any actual screw which we are investigating must be first of all increased by the 13½ per cent. or 20 per cent of the ellipse accounted for by the boss, and in using curves for converting nominal pitch to effective pitch an allowance should be made for the same reason. In other words, "B" values, "A" values, efficiency corrections, pitch-ratio corrections, and other constants depending upon blade area may be supposed to be based virtually upon mean-blade-width ratio. A propeller with a large boss has a greater mean-blade-width ratio for a given expanded-area ratio than has a propeller with a small boss, and in comparing and estimating the performances of the two propellers any constants which we use which depend upon area ratio should be those appropriate to the respective blade-width ratios.

Two propellers of identical diameter, pitch, and blade area, one with a large boss and the other having a small boss, are not so

like each other, so to speak, for purposes of comparison, as they would be if they had the same diameter and pitch and equal blade-width ratios—*i.e.* identical ratios of mean blade width to propeller diameter,—provided, of course, that the blade outlines are in as close resemblance as possible. Curves of Froude's "B" values may be plotted (1) for solid propellers with area ratios as abscissæ moved 13½ per cent. to the left, and (2) for built propellers with the abscissa scale of area ratios moved about 20 per cent. to the left, these modifications for actual blades giving higher values of the "B" constant than those tabulated in Mr Froude's 1908 paper for whole ellipses. The same modification applies to "A," which is merely $B \times p(p+21)$, and to efficiency correction.

TABLE XXXII.—"B" VALUES FOR SALT WATER.

Disc-area ratio.	·25.	·30.	·35.	·40.	·45.	·50.	·55.	·60.	·65.	·70.	·75.
Mr Froude's three blades, elliptical	..	·097 8	·102 0	·105 0	·107 0	·108 5	·110 0	·111 2	·112 4	·113 5	·114 7
Mr Froude's three blades, wide tip	..	·104 5	·109 7	·112 6	·114 8	·116 6	·118 2	·119 5	·120 7	·121 8	·123 0
Mr Froude's four blades, elliptical	..	·104 0	·110 6	·115 9	·119 7	·122 7	·124 9	·126 8	·128 2	·129 4	·130 6
Mr Taylor's three blades	..	·091 6	·095 3	·098 4	·101 2	·103 7	·106 1	·108 1	·110 0	·111 2	
Suggested values for Taylor's four blades	..	·097 5	·103 3	·108 6	·113 2	·117 1	·120 5	·123 1	·125 5	·127 0	

TABLE XXXIII.—VALUES OF "*a*" FOR TAYLOR'S THREE-BLADED PROPELLER IN SALT WATER.

Pitch ratio (p).	$p(p+21)$.	·25.	·30.	·35.	·40.	·45.	·50.	·55.	·60.	·65.	·70.	·75.
·6	12·96	..	1·188	1·235	1·275	1·312	1·342	1·377	1·401	1·425	1·441	
·7	15·2	..	1·392	1·449	1·496	1·54	1·576	1·615	1·646	1·672	1·691	
·8	17·43	..	1·599	1·661	1·716	1·768	1 809	1·851	1·889	1·919	1·94	
·9	19·7	..	1·805	1·878	1·94	1·997	2·04	2·093	2·132	2·168	2·192	
1·0	22	..	2·016	2·097	2·162	2·23	2·28	2·338	2·38	2·42	2·448	
1·1	24·33	.	2·28	2·32	2·395	2·465	2·522	2·585	2·636	2·679	2·71	
1·2	26·63	..	2·44	2·54	2·621	2·7	2·76	2·83	2·882	2·931	2·965	
1·3	29	..	2·66	2·762	2·852	2·94	3·005	3·08	3·139	3·19	3·23	
1·4	31·4	..	2·88	2·99	3·09	3·18	3·257	3·337	3·40	3·455	3·496	
1 5	33·8	..	3·1	3·22	3·323	3·425	3·504	3·59	3·66	3·72	3·762	
1·6	36·2	..	3·32	3·45	3·56	3·67	3·75	3·842	3·92	3·98	4·03	

TABLE XXXIV.—SUGGESTED VALUES OF "*a*" FOR TAYLOR'S FOUR-BLADED PROPELLERS IN SALT WATER.

Pitch ratio (p).	$p(p+21)$.	·25.	·30.	·35.	·40.	·45.	·50.	·55.	·60.	·65.	·70.	·75.
·6	12·96	..	1·348	1·432	1·50	1·55	1·59	1·618	1·64	1·661	1·678	
·7	15·2	..	1·58	1·681	1·76	1·82	1·865	1·898	1·925	1·95	1·969	
·8	17·43	..	1·812	1·929	2·02	2·087	2·14	2·176	2·21	2·239	2·258	
·9	19·7	..	2·05	2·18	2·281	2·36	2·418	2·458	2·497	2·53	2·55	
1·0	22	..	2·289	2·433	2·55	2·633	2·7	2·745	2·788	2·822	2·848	
1·1	24·33	..	2·531	2·691	2·82	2·915	2·985	3·04	3·081	3·122	3·15	
1·2	26·63	..	2·77	2·945	3·084	3·19	3·27	3·325	3·377	3·42	3·45	
1·3	29	..	3·016	3·21	3·36	3·47	3·56	3·62	3·675	3·72	3·755	
1·4	31·4	..	3·264	3·472	3·64	3·76	3·85	3·92	3·98	4·03	4·062	
1·5	33·8	..	3·516	3·74	3·917	4·05	4·15	4·22	4·285	4·34	4·38	
1·6	36·2	..	3·764	4·006	4·196	4·339	4·44	4·515	4·59	4·65	4·69	

SUMMARY OF METHOD FOR PROPELLER CALCULATION.

Use the curves for values of wake (Plate 66) by Mr Luke. Assume appendage factor and air resistance calculated from Taylor's KAV^2.

Use Froude's 1908 propeller efficiencies (Plates 49–51), based upon effective pitch from some diagram like Tobin's 1916 I.N.A., in which curves for different projected area ratios crossed by lines representing various B.T.F. plotted to a base of N.P.R. give a scale of factors for conversion of nominal pitch to effective pitch far coarser than 1·02, and more in keeping with blades with edges blunted by corrosion and ships with rough paint and shells. 1·02 may do for brand-new clean hulls and shining bronze blades, but even then it should be borne in mind that in model experiments the propellers run in open water, while in actual ships the wake is more or less disturbed, *i.e.* moving past the stern in an undefined way. The want of homogeneity in the wake has probably something to do with the difference in efficiency between model propellers and full-sized propellers.

Too low a value of the wake should not be taken, because it gives a speed of advance to work from with which maximum pressure in the engine is reached before there are sufficient revolutions per minute to yield the necessary power.

ROUGHER METHODS OF DETERMINING PROPELLER DIMENSIONS.

The types of propellers found in merchant-ship practice do not depart widely from a standard type, and the experience of

superintendent engineers with this type entitles them to some claim for an empirical method as one of the three main classes of methods,—all (it should be remembered) empirical at some stage. In rough methods of propeller design, wake is frequently not taken account of.

A formula such as the following,

$$K = \frac{D^2 \times \left(\frac{P \times R}{101 \cdot 33}\right)^3}{\text{I.H.P.}},$$

where D = diameter in feet,
P = pitch in feet,
R = revolutions per minute,
K = a constant,

may be turned to very good account if used continually by one who has a large collection of indicator diagrams and speeds and revolutions from actual service, and, with correct values of K taken from actual performances, it may be as useful as the formula for speed and power,

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}}$$

The propeller formula given above bears a close resemblance to Durand's formula,

$$U = (pN)^3 \times d^2 \times klm$$

where pN = pitch × revolutions.
d = diameter.
klm = constants.
U = thrust horse-power.

The only difference is that k is substituted for the three constants k, l, m.

As a check, the following expressions are useful :—

$$\left.\frac{\text{Projected area} \times V^3}{\text{I.H.P.}}\right\} \text{ where V = speed of ship in knots}$$

and

$$\left.\frac{\text{Indicated thrust in lbs.}}{\text{Projected area in square inches}}\right\} \text{ (an expression used by Captain Dyson)}$$

when taken in conjunction with

$$\frac{\text{Disc area} \times V^3}{\text{I.H.P.}}$$

$\dfrac{\text{Projected area}}{\text{Disc area}}$ and $\dfrac{\text{Projected area}}{\text{Expanded area}}$ are as useful and important for comparing and estimating the principal dimensions of a propeller as they are in calculations for blade thickness.

Plate 43 shows ratios of projected area ÷ expanded area for various area ratios.

PROJECTED AREA.

The expressions $\dfrac{\text{Projected area} \times V^3}{\text{I.H.P.}}$ and $\dfrac{\text{Indicated thrust in lbs.}}{\text{Projected area in sq. in.}}$ (a figure employed by Captain Dyson), when taken in conjunction with $\dfrac{\text{Disc area} \times V^3}{\text{I.H.P.}}$, are almost as important and useful coefficients as the empirical expression $\dfrac{(\text{Displacement})^{\frac{2}{3}} \times V^3}{\text{I.H.P.}}$.

Taylor gives the following expression:—If a = pitch ÷ diameter, projected area ÷ developed area $= 1{\cdot}067 - {\cdot}229a$ for his standard blade, of which the outline is a little fuller at the tip than the ordinary ellipse (see Plate 43), and the ratio $\dfrac{\text{Diameter of boss}}{\text{Diameter of propeller}} = {\cdot}20$.

Other blade shapes and boss diameters require other expressions for the ratio of projected area to developed area. A similar expression for some average blades having an outline like that of a man's thumb, where the diameter of boss ÷ diameter of propeller $= {\cdot}23$, is $\dfrac{\text{Projected area}}{\text{Expanded area}} = 1{\cdot}06 - {\cdot}204a$.

If the blade is raked aft, the expression becomes $(1{\cdot}06 - {\cdot}204a)$ sec α, α being the angle of rake if the blade is raked.

Taking the mean width of blade as 1·00, the widths may be figured on the contour in terms of the mean width. The widths for the projected area are calculated from the cosines of the pitch angles, and the respective areas calculated by summing and averaging the widths. The values of $\dfrac{\text{Projected area}}{\text{Expanded area}}$ may be plotted as ordinates of a curve on pitch ratios as abscissæ. Between pitch ratios ·90 to 1·6 the line is straight.

For merchant-ship blades of the following proportions,

Taylor's mean width = ·246,
Froude's width ratio = ·492,

the ratio $\frac{\text{Projected area}}{\text{Expanded area}} = 1{\cdot}064 - {\cdot}2a$ for ordinary pitch ratios up to 1·5.

The above apply fairly accurately to blades approximating in shape to the cubic ellipse, the equation for which is $\frac{x^3}{a^3} + \frac{y^3}{b^3}$.

Examples giving some Values of K from Actual Practice.

1. For a 9-knot single-screw cargo steamer, with D = 16 ft. 9 in., P = 16 ft. 6 in., R = 68, surface ratio = 0·30, K = 280, when the ship is loaded; and K = about 330, when the ship is light. (Roughly.)

2. For an 11-knot single-screw steamer about 300 ft. long, when $\frac{P}{D}$ = about 1·1 to 1·2 and R = about 80, K = about 310 when loaded. With $\frac{P}{D}$ = about 1·0, K = about 280; with $\frac{P}{D}$ = about 0·95, K = about 250.

3. For the torpedo-boat destroyer "Biddle," 30 knots at 325·2 revolutions. I.H.P. = 4 225, K = 443. (Twin screw.)

For the same T.B.D. at 20 knots, R = 220, K = 420; also "Biddle" at 25 knots, R = 273, K = 415.

4. The cruiser "Diadem." D = 16 ft. 9 in., P = 22 ft. 11⅛ in., expanded surface = 58; at 20·6 knots, R = 119·1, I.H.P. = 17 262, K = 313.

5. For our 460-ft. T.S.S. (see Plate 25) at full speed with $\frac{P}{D}$ = 1·24, area ratio = 0·321, K = 419.

6. The U.S. battleship "New Jersey," K = 294.

7. The U.S. battleship "Georgia," K = 327.

8. A 500-ft. twin-screw Atlantic liner. Block coefficient = 0·728, 16½ knots, 90 revolutions. $\frac{P}{D}$ = 1·25, area ratio = 0·32, K = 450. 100-ft. model $100 \times 11{\cdot}7 \times 5{\cdot}1$.

9. For an 18½-knot twin-screw steamer, 150 revolutions. Area ratio = 0·41, $\frac{P}{D}$ = 1·22, K = 410. 100-ft. model $100 \times 12{\cdot}8 \times 4{\cdot}5$.

10. For the U.S.S. "St Louis," $424 \times 66 \times 22{\cdot}6$ ft. mean draught. Displacement = 9 663. (Model $100 \times 15{\cdot}58 \times 5{\cdot}31$.) 22·13 knots, cylinders $\frac{36 \text{ in.} - 59\frac{1}{2} \text{ in.} - 69 \text{ in.} - 69 \text{ in.}}{45 \text{ in.}}$ 150 78 revolutions.

TABLE XXXV.—PROPELLERS: SMALL STEAMERS.

Name.	Δ.	Length.	Beam.	Mean draught.	Block coefficient.	I.H.P.	Knots.	Revolutions.	Propeller.				
									Diam.	Pitch.	Exp. area.	No. of blades.	$\dfrac{D^2\left(\frac{PR}{101\cdot33}\right)^3}{\text{I.H.P.}}$
T.S.S. Guardian	222	104·5	20	7·75	·48	187	9·94	102·8	7·33	11·0	23·9	4	$\frac{53\cdot9\times1\,380}{187}=397$
S.S. Argus .	406	188	23	7·5	·439	605	14	173	7·5	9·25	16	..	$\frac{56\cdot3\times3\,940}{605}=367$
S.S. Edgewater	687	173	34	9·8	·417	615	11·4	141	8·0	10·19	31·9	..	$\frac{64\times2\,850}{615}=298$
S.S. Manning .	1 000·7	188	32·81	12·33	·48	1 245	14	127·7	11·0	12·33	40	..	$\frac{121\times3\,780}{1\,245}=367$
T.B.D. Biddle .	168	157	16·25	4·81	·478	800	21	231·4	6·68	10·88	..	..	$\frac{44\cdot5\times1\,530}{800}=850$
Tug Iwana .	198	92·5	20·95	8·16	·438	349	11·58	115·5	7·5	12·5	22·5	4	$\frac{56\cdot3\times2\,880}{349}=465$
Tug Narkeeta .	190	92·5	20·95	7·92	·433	356	11·22	111·8	7·5	12·5	22·5	4	$\frac{56\cdot3\times2\,613}{356}=411$
Tug Wahneta .	176·5	92·5	20·95	7·6	·419	378	11·63	114·6	7·5	12·5	22·5	4	$\frac{56\cdot3\times2\,840}{378}=422$
Coasting S.S. .	520	155	28	6·33	·662	350	9·2	128	6·25	12·0	17·25	4	$\frac{39\times3\,470}{350}=387$
Screw steamer .	380	130	23	9·25	·482	650	11·75	225	7·5	6·75	..	4	$\frac{56\cdot3\times3\,350}{650}=290$
Gresham . .	820	188	32	9·92	·481	2 347	17·32	165·3	10·0	12·5	40	4	$\frac{100\times8\,400}{2\,347}=360$
S.S. —— . .	1 915	180·7	33·15	14·75	·76	600	9·0	95	11·0	10·5	44	4	$\frac{15-25-40}{24}$ Engines × 180 lbs. One boiler 14 ft. dia. × 11. Receiver pressures 50 lbs. and 10 lbs. 25-in. vacuum. Cut off. H.P. 19½-in. of stroke.

13 473 I.H.P., each screw three blades. D = 18, P = 19 ft. $0\frac{1}{16}$ in., area ratio = 0·36, L.W.L. coefficient = 0·67, prismatic coefficient = 0·61, mid-area coefficient = 0·87, wetted surface = 31 838. 44·92 tons per in. Transverse metacentre 14 ft. above C.B. K = 540.

In using Professor Durand's method of calculating screw-propeller dimensions, a slip ratio has to be definitely selected to work from. The slip ratio is involved by assuming a diameter and pitch ratio, or a diameter and different pitches, for trial of the method. If impracticable screw dimensions are produced, then a modified set of conditions must be assumed and the method applied again. All of the approved methods of screw-propeller design depend very much upon wake estimate. This is one reason why, at the present stage of research, the approved methods of calculation should be used with great caution for single screws. Tank experiments to ascertain the wake values, and interaction of hull and propeller, in single-screw cargo vessels, are very much to be desired. Testing model propellers separately, without reference to the model of the ship they are intended to drive, is of very little use. In tank research work, experiments are made upon the ship model without the propeller, upon the propeller apart from the ship, and upon the model ship with propeller behind it.

Slip ratio = slip per cent. divided by 100.

Let S = apparent slip per cent.
V = speed of ship in knots.
N = revolutions per minute.

Then

$$\text{Pitch} = \frac{V \times 101{\cdot}33 \times 100}{N(100 - S)}.$$

Mr T. S. Cockrill* gives a convenient formula for pitch of propellers as a guide in roughing out a design, generally within 2 per cent. of the most efficient propellers for all normal vessels. The actual dimensions for propellers can be determined in the later stages of the design.

Pitch of propellers in feet

$$= \frac{C \times K}{R},$$

where K = speed of vessel in knots,
R = revolutions per minute,
C = constant from following table :—

* *The Engineer*, 14th April 1916.

Type of vessel.	C.
9- and 10-knot cargo	109
12- and 13-knot cargo	111
Small naval (various)	114
Mail and intermediate liners	116
Cross-channel	120
Yachts, tugs, ferry-boats, etc.	124
Launches	140

Rake of blades, or "set back," should not be given to the blades when the propellers are very fast-running, because of the centrifugal stresses. Generally speaking, rake does not affect the efficiency, but keeps the blade tips at a proper distance from the hull in the case of wing screws without unduly spreading the shaft centres, and gives a better clearance between the leading edges of the blades and the stern-post or shaft struts.

In merchant ships a rake of from 5 to 8 degrees is given in the majority of cases.

"Skew back," or curvature of the blade in the transverse plane, is said to have no effect on the efficiency of the screw, but many superintendent engineers prefer to give a little skew or "throw-round" to help the propeller blade to clear itself of small obstructions in the water sometimes, and perhaps to minimise the shock when a blade is behind a thick web or stern-post.

Sometimes propellers are given a pitch which increases by about 10 per cent. from root to tip, with the idea of moderating the pitch angle at the root to give more thrust and less slip and less churning effect. It is very doubtful if anything is gained by this feature, which perhaps had its origin in an attempt to provide variable distribution of slip over the surface of the blade, "assuming the propeller to work in a uniform stream," which it does not. Plate 48 shows a graphic method of arriving at the effective face pitch of such a blade.

Various arrangements are made to break up synchronism of vibration in twin screws, such as three blades in one propeller and four blades in the other; or, more frequently, making one propeller rotate three or four revolutions per minute faster than the other.

Tug propellers are frequently made with very wide-tipped blades, which are less efficient when cruising than those with well-rounded tips, but this sacrifice is justified for the sake of the result when towing. Moderately small pitch ratios give the greatest pull when the boat is nearly stationary, perhaps 1·0 to 1·1 being the best, while a projected area ratio of ·50 as a maximum is recommended, with roughly about 120 revolutions per minute. Towing

with a long tow-rope, or with the vessel alongside when the water is smooth, is better than towing with a short tow-rope. A 300-ft. cargo steamer may tow well with only 6 per cent. extra coal consumption.

Practical Method of comparing Blade Stresses and calculating Root Thickness.

Referring to Plate 48, the blade may be treated as a cantilever; the cross section of the blade at the root, just where the generating line ends and the fillet begins, has a width $= b$ and a thickness $= h$. The modulus of section may be taken as

$$z = \frac{bh^2}{13}.$$

The length of the blade proper is measured on the longitudinal section of the blade through the line of greatest thickness, which is usually, though not always, a straight line. If the blade is one that has "throw round" in a transverse plain, something like a boomerang, its length may be measured along the curved line of greatest thickness from root to tip.

Referring to Plate 48, the load on the blade may be supposed to be applied to the centre of pressure, and the length of the arm of the cantilever measured from the root to the centre of pressure may be taken as $\frac{3}{5}$ or ·6 of the blade length, if the blade is of an ordinary shape.

The delivered thrust per blade, W

$$= \frac{\text{D.H.P.} \times 33\,000}{\text{Pitch} \times \text{revolutions per minute} \times \text{number of blades}}.$$

The formula for a cantilever, $Wl = fz$, may now be applied. The ratio $\frac{\text{Expanded area}}{\text{Projected area}}$ is a function of the pitch angle.

Here we have

$$f = \frac{W \times l \times \text{Expanded area}}{z \times \text{Projected area}},$$

where $f =$ the stress in lbs. per square inch at the root, h.
$W =$ the delivered thrust in lbs. on each blade.
$l =$ the length of the arm $=$ ·6 blade length.*
$z =$ the modulus of section at the root, $\frac{bh^2}{13}$.

* With blades of abnormal width at the tip, the "arm" might be increased somewhat.

For cast iron, f = about 2 800.

For cast steel and manganese bronze, f = about 5 500.

When the fillet at the root of the blade connecting the blade to the flange (or to the boss in the case of a solid propeller) is of very large radius, perhaps a slightly higher stress f may be allowed if desired than in cases where the radius is small.

For instance, for a blade set back a foot at the tip, having a skew back of about 8 degrees, 6 ins. would be an average radius for the driving face, and about 11 ins. for the radius at the back. A blade connecting with the usual radii to a flat flange is at a disadvantage in strength, compared with a blade having a flange shaped as if to form part of a spherical boss, even though the radii in the two cases are the same. If the flange must be flat, the radii should be increased. Bronze blades tend to twist to coarser pitch in the course of their work, and for this reason they are often made as thick as they would have to be if cast steel were the material employed. The late Mr Blechynden mentioned the springing of bronze blades in a paper to the North-East Coast Institution of Engineers and Shipbuilders, and the author has evidence of it with a large passenger steamer driven in rough weather; the pitch of the propellers measured afterwards, however, was not greater than the original. Professor Durand speaks of it as a bending of the blade as a whole under the influence of the thrust, the bending being accompanied by a slight untwisting of the blade, thus tending toward an increase of effective pitch and slip so as to sensibly affect the efficiency, usually for the worse.*

Most authorities state that good cast-steel propellers can be given the same stresses as those of manganese bronze. We should rather say, for merchant steamers, assume the cast steel only moderately good in quality, and make the blades strong but not too thick; then, if bronze blades are substituted for the cast-steel blades, make them of the same thickness as the cast steel, to avoid springing. As cast-iron blades do not bend and are apt to be brittle, they are necessarily thicker and therefore less efficient than those of steel or bronze; but a solid propeller of cast iron, with a small clean boss, works with less eddying than a built propeller, and often lasts twice as long as a set of steel blades. The latter are usually wasted by corrosion after two and a half years' work. The line of greatest thickness is usually at the middle of the width of the blade, *i.e.* h is a maximum at a distance

* Mr Taylor mentions in his book a vessel which much exceeded her designed power on trial, and also sprung her propeller blades. This may mean a permanent distortion of the blades, but our remarks refer to temporary springing.

$\frac{b}{2}$ from the leading edge, the back of the blade being drawn an arc of a circle. Propellers in air show a gain in efficiency and in thrust by moving the maximum thickness to about ·38b from the leading edge (not nearer).

This design seems to give good results in water, but it is not certain that it is any better than the symmetrical ogival section.

The same stress (f) and modulus of section (z) might be taken for either shape.

Plate 45 shows that the thicker and narrower the blades the more the virtual pitch is increased as compared with the nominal or face pitch (the finer the pitch ratio the greater this difference); and the same is true of the slip, at least up to pitch-ratio unity, above which, if the blades are narrow and over a certain thickness, the back of the blade—to use Mr Taylor's expression—begins to lose its grip of the water and the increase of effective pitch over nominal face pitch is less marked.

$\frac{\text{Expanded area}}{\text{Projected area}}$ is the pitch-angle factor.

When the blade has a skew back of α degrees, the ratio of the projected area to the expanded area is diminished by the cosine of the angle of skew back ($\cos \alpha$); but as we have to consider the length of the driving face, we should have to correct the pitch-angle factor by taking the reciprocal of the cosine. The formula then becomes

$$f = \frac{Wl}{z} \times \frac{\text{Expanded area}}{\text{Projected area}} \times \sec \alpha,$$

strictly speaking.

Calculation of Blade Strength, Root Thickness, and Strength of Bolts or Studs securing Blade Flange to Boss.

The usual custom of taking the I.H.P. instead of the D.H.P. (horse-power delivered to the propeller) is quite in order in comparisons and for drawing-office calculations for steamships driven by reciprocating engines. The S.H.P. is perhaps better, and D.H.P. better still.

By means of the curves on Plate 40, the D.H.P. can be obtained from the I.H.P. for any ordinary engine, and there is no reason why D.H.P. should not be always used.

Example.—Single screw, four blades, cast steel. Diameter = 16 ft. 9 in. Pitch = 17 ft. 6 in. Expanded area = 84 sq. ft. Projected area = 70.

From indicator diagrams we have the following data of I.H.P. and revolutions :—

I.H.P.	Revolutions per minute.	Indicated thrust per blade in lbs.
1 771	72·5	11 520
2 151	74·5	13 620
2 054	73	13 260
2 038	72	13 340
1 801	64	13 270
2 049	73·5	13 130
2 065	71	13 720
1 979	70	13 320

By inspection, a good average seems to be 2 065 I.H.P. at 71 revolutions.

Lbs. indicated thrust per blade (W) = 13 720.

$$\frac{\text{D.H.P.}}{\text{I.H.P.}} = {\cdot}856 \times {\cdot}97 = {\cdot}83 \text{ (from Plate 41)}.$$

∴ Delivered thrust per blade = 11 400 lbs.

The blades are nearly elliptical, and their breadth at root, where the blade proper joins the radius or fillet to the flange, = 36 in. The thickness is $6\frac{1}{2}$ in. $= h$, $b = 36$.

The length of the blade proper is 75·5 in.

·6 × length of blade = 45·2 in., = (l) length of arm for load.

$$z = \frac{bh^2}{13} = \frac{36 \times (6\frac{1}{2})^2}{13} = 117.$$

$$f = \frac{Wl}{z} \times \frac{\text{Expanded area}}{\text{Projected area}}$$

$$= \frac{11\,400 \times 45{\cdot}2 \times 84}{117 \times 70} = 5\,300 \text{ lbs. per square inch stress at root.}$$

Cast-steel blades of these proportions worked satisfactorily on a pair of steamers for a number of years. Thinner blades cracked, and thicker blades reduced the ship speed.

Each blade is secured to the boss by seven studs—four on the driving side. Let the average stress on the studs of the driving

side $= f$. The flange diameter = the average leverage of the bolts from their fulcrum on the opposite side of the flange = about $25\frac{1}{2}$ in.

$$f = \frac{\text{Delivered thrust per blade in lbs.} \times \text{effective leverage of blade in inches} \times \text{expanded area}}{\text{Bolt leverage in inches} \times \text{number of bolts on driving face} \times \text{tension area of one bolt} \times \text{projected area}}$$

$$= \frac{11\,400 \times 60{\cdot}5 \times 84}{25{\cdot}5 \times 4 \times 6{\cdot}1 \times 70} = 1\,330 \text{ lbs. stress on bolts or studs per sq. inch.}$$

(This is a very moderate stress.)

The usual drawing-office custom of taking the I.H.P. or S.H.P. instead of the D.H.P. (delivered horse-power), is quite in order for comparisons and rough calculations.

In the above example

$$\text{I.H.P.} = 2\,065.$$

Indicated thrust per blade

$$= \frac{2\,065 \times 33\,000}{17{\cdot}5 \times 71 \times 4} = 13\,720 \text{ lbs.} = \text{W}.$$

$$f = \frac{\text{W}l}{z} \times \frac{\text{Expanded area}}{\text{Projected area}}$$

$$= \frac{13\,720 \times 45{\cdot}2}{z} \times \frac{84}{70}$$

$$= \frac{13\,720 \times 45{\cdot}2}{117} \times \frac{84}{70}$$

$$= 6\,370 \text{ lbs. per square inch.}$$

Cast-steel blades of these proportions worked well, as stated above, for many years. Slight alterations in the thickness were never attended with success: thicker blades were less efficient, and thinner blades broke. Thus we have, based upon D.H.P., 5 300 lbs. stress, and based upon I.H.P. 6 370 lbs. stress. It does not matter which we base it upon so long as we use the corresponding figure, but of course it is usual to stick to one basis of comparison.

Example.—Let us examine the stress f at the root of the propeller blades of the Hamburg-American T. S. steamer "Kronprinzessin Cecilie," illustrated in *International Marine Engineering*, January 1908. Four manganese bronze blades. 79 revolutions

per minute. Diameter = 17 ft. 0¾ in. Pitch = 20 ft. 4⅛ in. At a radius of 30⅝ in. from the centre of the shaft the width of a blade is 37·5 in., the blade proper is just touching the fillet or radius to the flange, and the thickness of the root section at that part is 6¼ in.

$$z = \frac{bh^2}{13} = \frac{37 \cdot 5 \times (6\frac{1}{4})^2}{13} = 112 \cdot 5.$$

$$l = \cdot 6 \times \text{length of blade proper} = \cdot 6 \times 73\tfrac{1}{2} \text{ in.} = 44 \cdot 1 \text{ in.}$$

Indicated horse-power each propeller = 3 035.

Take

$$\frac{\text{D.H.P.}}{\text{I.H.P.}} = \cdot 84. \quad \therefore \text{D.H.P.} = 2\,550.$$

Then the delivered thrust per blade

$$= \frac{2\,550 \times 33\,000}{20 \cdot 344 \times 79 \times 4}$$

$$= 13\,100 \text{ lbs.}$$

$$f = \frac{Wl}{z} \times \frac{\text{Expanded area}}{\text{Projected area}}$$

$$= \frac{13\,100 \times 44 \cdot 1}{112 \cdot 5} \times \frac{86 \cdot 5}{69 \cdot 4}$$

$$= 6\,390 \text{ lbs. per square inch.}$$

The springing of the manganese bronze blades of the four-bladed propeller of the twin-screw passenger steamer referred to on p. 185 occurred when the root thickness gave a stress of 7 800 lbs. per square inch, but it is probable that the springing was due to the fact that the blades were "hollow-backed," as on the right-hand sketch on Plate 48. The upper part of the blade in that case lends itself to this action. When thicker blades were fitted, giving a stress at the root of 7 200 lbs. per square inch, the blades being "straight-backed," there was an increase of 2½ revolutions per minute, and a corresponding improvement of the ship's speed. In the same fleet a cargo steamer used one set of manganese bronze blades continuously for over twenty years without change, the blades being much thinner than the average practice. The stress at the root worked out at 8 100 lbs. per

square inch. Possibly there was some springing, but there was no opportunity of comparing the revolutions and speed with those which would have been obtained by the substitution of thicker blades, and the propeller suited the ship very well. Two cast-steel spare blades were carried on board during the life of the ship, but they were never used.

When blades are made with a flange shaped to form part of the sphere of the boss, and the blades are recessed into the boss, great care should be taken to have the flange periphery turned to gauge slightly smaller than the recess, and to see that the blade flange is properly bottomed when bolted on, otherwise there is trouble with loosening of nuts, snapping of studs, and sometimes snapping of the blade at the root or throwing off the blades. A flange bolted on the outside of a flat face on the boss gives least mechanical trouble, though it makes the propeller angular at the hub.

If the working blades are bronze, the blade studs are usually of insufficient length to take the thicker flange which cast-iron blades would have if these were supplied as spares. In approving drawings of new propellers, therefore, owners may in some cases ask for the blade studs and nut facings to be deepened to suit possible cast-iron spare blades. Steel blades have the same flange thickness as bronze. With bronze blades the edges remain sharp and the surfaces smooth, and a steadier speed of ship is maintained through successive voyages than when using cast-steel or cast-iron blades which are liable to corrosion. Steel blades corrode quickly; in a few months the edges become blunt, and the surfaces rough and deeply pitted. Their average life is two and a half years.

In cargo steamers, where the draught varies considerably, and the blades are exposed to the action of air, cast-iron and cast-steel blades waste rapidly at the tips. A cast-iron solid propeller is often very efficient for some time, but when steel and iron blades become blunted and roughened by corrosion some months before they are renewed, the efficiency must be very low. In collecting data from performances of blades which are liable to heavy corrosion, a mean should be taken from results, first with the blades in good condition, and then from log abstracts with the blades in the blunt and rough condition.

Tug "Arary." Single screw. 7 ft. diameter × 8 ft. 9 in. pitch. Four blades. Cast iron loose. Expanded area = 22 sq. ft. Projected area = 18·5. Engines $\frac{10\frac{1}{4}-16-26}{20}$ × 185 lbs. W.P. 140 revolutions.

Take 230 I.H.P. at 140 revolutions. Take D.H.P. = ·81 I.H.P. = 187.

$$\text{Delivered thrust per blade} = \frac{\text{D.H.P.} \times 33\,000}{8{\cdot}75 \times 140 \times 4} = 1\,260 \text{ lbs.}$$

The net length of the blade from tip to the beginning of the root fillet is $29\frac{1}{2}$ in.

The load of 1 345 lbs. may be supposed to be concentrated upon the centre of pressure, say ·6 × length of blade, measuring from the root, *i.e.* at 17·7 in. from the root section, where the radius commences to connect blade with flange.

17·7 in. = the effective "arm" of the cantilever.

The transverse section of the root is $17\frac{1}{2}$ in. wide × $2\frac{13}{16}$ in. thick.

$$b = 17{\cdot}5.$$

$$h = 2{\cdot}812\,5.$$

$$z = \frac{bh^2}{13} = 10{\cdot}7.$$

$$f = \frac{Wl}{z} \times \frac{\text{Expanded area}}{\text{Projected area}} = \frac{1\,260 \times 17{\cdot}7}{10{\cdot}7} \times \frac{22}{18{\cdot}5}$$

$$= \frac{1\,260 \times 17{\cdot}7 \times 22}{10{\cdot}7 \times 18{\cdot}5} = 2\,480 \text{ lbs. per square inch.}$$

Each blade is secured to the boss by four studs, two on the driving face. Let the stress per square inch on the two studs of the driving face $= f$. The flange is $13\frac{1}{2}$ in. diameter, and the bolts are $10\frac{5}{8}$ in. from their fulcrum on the opposite side of the flange.

$$f = \frac{\text{Delivered thrust per blade lbs.} \times \text{effective blade arm in inches} \times \text{expanded area}}{\text{Bolt leverage in inches} \times \text{number of bolts on driving face} \times \text{tension area of one bolt} \times \text{projected area}}$$

$$= \frac{1\,260 \times 17{\cdot}7 \times 22}{10{\cdot}625 \times 2 \times 1{\cdot}30 \times 18{\cdot}5} = 961 \text{ lbs. per square inch tension.}$$

Both of these values of f are moderate.

The following formulæ were given by Mr T. Sidney Cockrill to the Liverpool Engineering Society in 1906 :—

(1) Thickness of blades at the root :

$$\frac{\text{BH}^2 \times \textbf{N} \times \text{P}}{\dfrac{\text{I.H.P.}}{\text{No. of blades}} \times (\text{D} - d)} = \text{C},$$

where B = breadth of blade at root in inches.
H = thickness of blade at root in inches.
N = revolutions per minute.
P = pitch in feet.
D = diameter in feet.
d = diameter at root in feet.
I.H.P. = power of engine driving each propeller.
C = 130 for manganese bronze.
175 for cast steel.
225 for gun-metal.
500 for cast iron.

(2) The thickness at tip in inches from the following table :—

	Cast iron.	Cast steel.	Gun-metal.	High-class bronze.
For propeller 7 ft. diameter	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{3}{8}$
,, ,, 13 ,, ,,	$\frac{15}{16}$	$\frac{13}{16}$	$\frac{9}{16}$	$\frac{1}{2}$
,, ,, 19 ,, ,,	$1\frac{3}{16}$	1	$\frac{3}{4}$	$\frac{5}{8}$

(3) Size of studs or bolts for securing loose blades to boss :—

$$a \times N \times r = \frac{T \times L}{K}.$$

where a = area of one stud or bolt at bottom of thread in square inches.
N = number of studs or bolts for one blade, usually 7, 9, or 11.
r = radius of pitch circle of studs in inches.
T = indicated thrust per blade.
L = ·6 × total length of blade (flange joint to tip) in inches.
K = 1 700 for mild-steel studs.
1 400 for forged bronze or naval bronze studs.

Durand gives the following expression for the thickness of propellerblades :—

$$T = A\sqrt{\frac{He}{bNn}},$$

where

$p \div d$.	e.	$p \div d$.	e.
1	1·10	1·7	·74
1·1	1·02	1·8	·72
1·2	·95	1·9	·70
1·3	·89	2·0	·68
1·4	·85	2·1	·66
1·5	·81	2·2	·64
1·6	·77	2·3	·63

t = thickness in inches at root of blade, *i.e.* where the blade intersects the hub (the fillet by which it is connected to the hub is extra, and is not here considered).

H = I.H.P.

e = factor from table above.

b = length in inches of section at root of blade.

N = revolutions per minute.

n = number of blades.

$A = \begin{cases} 9 \text{ to } 12 \text{ for bronze or steel.} \\ 14 \text{ to } 17 \text{ for cast iron.} \end{cases}$

In Taylor's figures the thickness of blade is produced to the shaft centre line, CA being the distance measured, at the shaft, between the face and back lines of the blade produced, *i.e.* the axial thickness. Then $\frac{\text{CA}}{\text{Diameter of propeller}}$ = blade-thickness fraction.

For a merchant ship with four-bladed propeller, with cast-steel blades, ·042 is an average blade-thickness fraction; for example, 8¼ inches axial thickness (measured along the shaft), with diameter = 16 ft. 9 in., and pitch = 21 ft., revolutions = 74. For bronze = ·04. For cast-iron solid propellers for the same type of vessel the blade-thickness fraction is about ·051 to ·055, which is nearer Taylor's standard for the area ratios usually adopted, viz. about ·38 for four-bladed, and about ·35 for three-bladed, propellers, corresponding to mean-width ratio of ·25.

CHAPTER IX.

MISCELLANEOUS DATA.

NOTED from paper read before joint meeting of N.E. Coast Inst. Engineers and Shipbuilders, and Inst. Engineers and Shipbuilders, Scotland, 4th August 1908, by Engineer-Commander Wisnom, R.N.

S.S. "Otaki," built in 1908 by Messrs Denny, Dumbarton, for the New Zealand Shipping Co. Designed for a continuous sea speed of 12 knots when fully loaded, and 14 knots with 5 000 tons deadweight on trial. Length, b.p. 465·4 × 60·3 × 31·3. 9 900 tons deadweight on a draught of 27 ft. 6 in. Block coefficient = about ·757. Three shafts. Engines: Two sets reciprocating, driving the wing propellers, and a low-pressure Parsons turbine driving a centre propeller.

Cylinders $\frac{24\frac{1}{2}\text{ in.} - 39\text{ in.} - 58\text{ in.}}{39\text{ in.}} \times 200$ lbs. W.P. Five S.E. boilers. G.S. = 305 sq. ft. Total H.S. = 13 500 sq. ft.

Howden's F.D. Turbine rotor drum = 7 ft. 6 in. diameter. Lengths of blades = $4\frac{3}{4}$ in. to $12\frac{11}{16}$ in. Two condensers. Total cooling surface = 6 000 sq. ft. (Contraflo). Total cooling surface = 6 000 sq. ft. Two 16-in. bore centrifugal pumps = 150 revolutions per minute. 48-in impellers.

Trial at Skelmorlie, 31st October 1908. Mean draught = 20 ft. 1 in. Displacement = 11 716 tons. Block coefficient = ·728.

The total feed water used for all the engines was measured by tanks during the trials. The water consumption as calculated from the number of strokes of the feed pumps was found to be in all cases greater than that obtained by the tank measurements, the difference being about 3 per cent. at the higher speeds. The results do not include make-up feed. The "Otaki" is virtually a sister ship to the twin-screw vessels "Orari" and "Opawa." The total horse-power of the "Otaki" was taken as the I.H.P. of

Total horse-power.	Speed in knots.	E.H.P.	Skin H.P.	Residuary H.P.	Revolutions per minute.			Lbs. mean absolute pressures.	
					Port.	Starboard.	Centre.	H.P. receiver.	Turbine inlet.
6 857	15·02	3 700	2 580	...	103	103·5	224·5	193	9·5
5 348	14·278	...	2 240	...	96·2	97·9	209·7	178	7·62
4 704	13·829	...	2 000	...	93·1	93·5	197·2	166	6·76
3 282	12·518	...	1 510	...	84·6	83·4	172·1	135	5·0
1 960	10·67	...	970						

the reciprocating engines plus S.H.P. of centre shaft. Scotch coal was used. Evaporation from and at 212° F. = 14 lbs. The water consumption per E.H.P. hour was found to show a gain of 20 per cent. in "Otaki," the propulsive coefficient of the reciprocating-engined "Orari" being ·60 at 14·6 knots as against ·57 in "Otaki" at the same speed.

T.S.S. "Orari," built in 1906 by Messrs Denny, Dumbarton, for the New Zealand Shipping Co.

See Commander Wisnom's paper read before the joint meeting of the N.E. Coast Inst. Engineers and Shipbuilders, and the Inst. Engineers and Shipbuilders in Scotland, 4th August 1908.

·60 ⇒ Propulsive coefficient. $\frac{B}{H} = 2·886$ at 20 ft. 1 in. draught.

$\frac{D}{\left(\frac{L}{100}\right)^3} = 119·7.$ $\frac{V}{\sqrt{L}} = ·68.$ $\frac{B}{H} = 2·54$ at 23 ft. 6 in.

Knots.	I.H.P.	E.H.P.
		At 23 ft. 6 in. mean draught.
14·6	5 360	3 210
14·31	5 000	
14·0	4 590	
13·65	4 200	
13·0	3 550	
12·29	3 000	
11·7	2 600	

Full-speed measured mile trials of Orient liners, 1909. All at 24 ft. 3 in. mean draught.

Name.	Tons displacement.	Lloyd's dimensions.	Revolutions per minute.	I.H.P.	Mean speed knots.	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}}$
Orsova .	15 160	536·2 × 63·3	85	11 700	18·1	310
Otway .	15 130	535·9 × 63·2	93	11 724	18·2	315
Osterley .	15 280	535·0 × 63·2	93·5	13 790	18·76	295
Otranto .	15 160	535·3 × 64·0	93	14 450	18·95	289

T.S.S. "Osterley." Progressive trial, 18th June 1909. Lloyd's length and beam. 535 × 63·2 × 24·25 mean draught. Block coefficient = ·653. Flat keel.

Cylinders $\frac{28\frac{3}{4}\text{ in.} - 41\text{ in.} - 58\frac{1}{2}\text{ in.} - 84\text{ in.}}{60\text{ in.}} \times 215$ lbs. W.P.

F.D. Heating surface = 31 038 sq. ft. Grate surface = 682 sq. ft. Four D.E.B. Two S.E.B.

None of the pumps were worked off the main engines, all were independent, including the air pumps.

Runs.	Mean revolutions.	Mean I.H.P.	Mean speed in knots.
1st. Up and down .	61·1	3 743	13·01
2nd. ,, ,, .	70·52	5 515·5	14·96
3rd. ,, ,, .	77·5	7 345	16·4
4th. ,, ,, .	83·3	9 403	17·23
5th. ,, ,, .	88·15	11 157	18·06
6th. ,, ,, .	93·5	13 790	18·76

19th June, Cloch to Cumbrae, four double runs, each of 13·66 miles.

Runs.	Mean revolutions.	Mean I.H.P.	Mean speed in knots.
Mean of four runs .	...	12 240	18·29

Mean draught as on service = 24 ft. 3 in. Displacement = 15 300 tons. 8 640 I.H.P. on consumption trials. Speed according to revolutions = 16¼ knots. 105 tons coal consumed in twenty-four hours.

"Otway," twenty-four hours' consumption trial, 24 ft. 3 in. draught, about 15 000 tons displacement, 17·16 knots mean speed, 9 170 I.H.P., 412 miles, 127 tons coal, 1·29 lbs. per I.H.P. hour.

"Lusitania," trial at 32 ft. 9 in. mean draught. Displacement = 37 080 tons.

Propulsive coefficient.	Speed in knots.	S.H.P.	Revolutions.	App. slip per cent.	Pressures, lbs.		Steam consumpt. per S.H.P. hour of turbines.		Total steam consumpt. in lbs. per S.H.P. hour.	Coal consumpt. in lbs. per S.H.P. hour.
					H.P. receiver.	L.P. receiver.	Main turbines.	Auxiliaries.		
·47	25·62	76 000	194·3	17·2	157	5¼	..	..	..	..
·475	25·4	68 850	..	..	..	..	12·77	1·69 (2·17)	14·46 (14·92)	1·43 (1·46)
·48	25·0	65 500	186	15·5	135	2½	..	..	..	..
·492	23·7	51 300	174·2	14·5	110	½	..	..	..	..
·50	23·0	48 000	..	..	..	..	13·92	2·01 (2·65)	15·93 (16·57)	1·56 (1·62)
·500 8	22·02	40 500	161·5	14·3	90	3½″ Vac.	..	..	..	..
·515	21·0	33 000	..	..	..	..	14·91	2·6 (3·41)	17·51 (18·32)	1·68 (1·8)
·501 9	20·4	29 500	147·6	13·1	70	6½″ Vac.	..	..	...	..
·50	18·0	20 500	131·1	13·7	50	10½″ Vac.	17·24	3·72 (4·92)	20·96 (22·16)	2·01 (2·17)
	15·77	13 400	116·1	14·6	35	14½″ Vac.	21·33	5·3 (6·97)	26·53 (28·2)	2·52 (2·76)

In the above trial the turbo-generators were exhausting to auxiliary condensers, other auxiliaries exhausting to heaters.

(The figures in brackets show the estimated figures for consumption under actual service conditions for the washing-water supply, etc., with a full complement of passengers, weather conditions being as on official trial.)

At 65 000 S.H.P. on voyage, the evaporating plant and heating took ·5 lb. steam per S.H.P. hour. Water evaporated per lb. of coal = 10·2 from feed temperature of 196°. Evaporation per lb. coal from and at 212° = 10·9 lbs. Coal per square foot of grate per hour = 24·1 lbs.

Triple-screw turbine steamers "Heliopolis" and "Cairo" (see *Engineering*, 24th January 1908). 525 ft. b.p. × 60·2 ft. beam.

Depth = 38 ft. keel to shelter deck. Gross tonnage = 12 000. 18 000 S.H.P. Mid-area coefficient = ·904. 180 lbs. boiler pressure.

"Heliopolis," 21·9 knots for three hours in the Irish Sea. Plymouth to Marseilles in 95½ hours. Marseilles to Alexandria in 72½ hours. 20·6 knots on twelve hours' trial at about 16 800 S.H.P., 340 revolutions full power.

"Heliopolis" at 21 ft. 5½ in. draught, 20·53 knots, 366·3 revolutions per minute.

"Cairo," 22 ft. draught, 20·6 knots, 372·5 revolutions. On trial, 18·35 knots, 10 800 S.H.P.

Revolutions.	Knots.
200	12·198
261	15·419
314	18·16
346	19·73
372	20·75

Danish royal yacht "Dannebrog" (paddle). Lengthened from 192 ft. to 227 ft. in 1907. 227 ft. b.p. × 26 ft. 2 in. mld. × 9 ft. 10 in. mean draught. Δ = 1 063 tons. Block coefficient = ·7. Oscillating engines, four hours' trial in 1907. Draught = 9 ft. 10½ in. Δ = 1 063 tons. 13·04 knots at 937 I.H.P. Apparent slip per cent. = 22·08.

Knots.	Revolutions per minute.	I.H.P.	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}}$.
8	18·0	210	254
9	20·25	276	275
10	22·6	380	274
11	25·0	536	264
12	27·5	725	248
13	30·0	937	244
13·2	30·4	990	242

I.H.P. varies as $V^{2 \cdot 32}$ between 8 and 9 knots.
" " $V^{3 \cdot 47}$ " 11 " 12 "
" " $V^{3 \cdot 2}$ " 12 " 13 "

French T.B.D. "Bouclier" (see *The Engineer*, 5th December 1911). Four propellers, three shafts. 233·33 × 24·83 (extreme) × 12·5 ft. mean draught. Displacement at trials, 660 tons. Contract speed, 31 knots. Trial speed, 35·334 knots. Beam, 10·64 per cent. of length. $\frac{L}{B} = 9{\cdot}4$. $\frac{B}{D} = 1{\cdot}988$. $\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 52$.

Parsons turbines, direct. Four Normand boilers with 5 277 sq. ft. heating surface. 228 lb. W.P.

Shaft horse-power measured by Hopkinson-Thring torsionmeter. One propeller on each shaft. 5 ft. 3 in. diameter × 4 ft. 11 in. pitch.

	Six hours' full-power trial.	Eight hours' consumption trial.
Displacement at start	650·44 tons	659·446 tons
Average mean steam pressure at boilers	217 lbs.	214 lbs.
,, ,, chest pressure .	183 lbs.	
,, air pressure in stokeholds .	4·28 in.	
Pressure in liquid-fuel burners per sq. in.	143 lbs.	
Revolutions per minute, mean . .	1 034·2	325·19
Mean speed for full time of trial . .	35·334 knots	14·06 knots
Contract speed	31 knots	14 knots
Shaft horse-power	15 000	1 400
Vacuum	27·4 in.	28·6 in.
Consumption of fuel per hour . .	21 912 lbs.	1 915 lbs.
,, per sq. ft. heating surface	1·038 lb.	
,, per shaft horse-power hour	1·46 lb.	1·37 lb. (nearly)
$\frac{\Delta^{\frac{2}{3}} V^3}{S.H.P.}$	221	151
$\frac{V}{\sqrt{L}}$	2·318	·923

Transactions American Society Naval Architects and Marine Engineers, 1911. Paper by W. L. R. Emmet, Esq., "Electrically-propelled Fire-boat, 'Graeme Stewart.'" L.W.L., 111 × 27 ft. 6 in. × 9 ft. draught to top of keel (dimensions scaled from a small drawing). Speed and power curves. Propeller, D = 6 ft. Pitch at tip, 6 ft. 9 in. Pitch at 9-in. radius = 5 ft. 9 in. Expanded surface = 16·6 sq. ft. Projected area = 13·75. Propeller of insufficient size; excessive slip at higher speeds.

Miles per hour, 5 280 ft.	Motors: total brake H.P.	Revolutions per minute.	Apparent slip per cent.
4·6	63	72	9·1
5·0	65	78	9·2
6·0	68	92·5	9·5
7·0	75	109	10
8·0	120	126	10·9
9·0	184	145	12·3
10·0	295	166	14·4
11·0	500	192	18·5

In this vessel General Electric Co.'s turbines drive centrifugal fire-pumps. These turbines are also connected to direct-current generators, each of the twin-screw propellers being driven by a motor.

"Vulcanus," built at Amsterdam, 196 ft. × 37 ft. 9 in. × 13 ft. 2½ in. Load draught, 10 ft. 2 in. Displacement about 1 900 tons (see *The Shipbuilder*, 1911, vol. vi., No. 21). Single-screw direct driven by Werkspoor oil-engine, reversible Diesel, 500 B.H.P. at 180 revolutions per minute. 8·4 knots (see below). Six cylinders, four-cycle. 15¾ in. diameter × 23⅝-in. stroke. Weight of complete installation of propelling machinery = 85 tons. Weight of engine alone = 42 tons.

A similar Diesel engine of 40 to 50 B.H.P. drives auxiliary machinery by compressed air.

A 10-H.P. Deutz electric-light engine.

The Shipbuilder, No. 22, gives the following particulars :—

Displacement. Tons.	Time. Days	hrs.	mins.	Distance nautical miles.	Fuel consumption in tons.	Speed in knots.	Fuel tons per 24 hours.	Lbs. fuel per B.H.P. per hour.
2 200		19	45	141	1·8	7·14	2·19	·409
1 480	19	4	15	3 263	37·5	7·1	1·956	·365
2 180	20	22	35	3 595	42·0	7·15	2·008	·374
1 360	1	19	0	360	2·0	8·38	1·115	·573

By another account the cylinders of the engine were 16·7 in· diameter × 23½-in. stroke. Six cylinders. 450 B.H.P. at 180 revolutions per minute. The four-stroke cycle. With 90 working strokes per minute, say 64 lbs. mean pressure per sq. in., we have

$$\frac{(\text{area of } 16{\cdot}7\text{-in. piston}) \times 1{\cdot}96 \text{ ft.} \times 90 \times 64}{33\,000} = 75 \text{ B.H.P. per cylinder.}$$

Motor ship "California," Burmeister & Wain, 1913 (see *The Engineer*, 10th October 1913). 405 ft. × 54 ft. × 23 ft. 3 in. draught for 11 000 tons displacement.

Two eight-cylinder Diesel engines similar to those of "Selandia." Cylinders 540 mm. bore × 730 mm. stroke, 2 700 combined I.H.P. at 140 revolutions. Well over 11 knots on ·38 lb. fuel per shaft H.P. hour, including auxiliary engines. Separate fuel pump for each cylinder. Reversing gear consists of a simple compressed air cylinder on the lines of the Brown steam gear.

Two three-cylinder 180 B.H.P. Diesel engines drive the dynamos and three-stage compressors; cargo winches driven by steam provided by an oil-fired boiler with 1 000 sq. ft. of heating surface. The windlass is electrically driven. Hele-Shaw electrical steering system.

Twin-screw U.S. T.B.D. "Balch" (from the *Journal of the American Society of Naval Architects*, and *The Shipbuilding and Shipping Record*, 26th August 1915). 300 ft. l.w.l. × 30·33 ft. at l.w.l. × 9 ft. $2\frac{1}{2}$ in. $\Delta = 1\,010$ tons. Tons per inch = 14·21. Area immersed midship section, 190 sq. ft. Coefficient = ·68. Block coefficient = ·415. Prismatic coefficient = ·611.

Cramp-Zoelly turbines combined with reciprocating engines. Cruising engine $\frac{13-25}{12}$ with cranks at 180°, 300 revolutions for speed of $15\frac{1}{2}$ to 16 knots.

PROGRESSIVE TRIALS.

	Four hours' full-power trial.	Four hours' 24-knot trial.	Four hours' $15\frac{1}{2}$-knot trial.	Four hours' 12-knot trial.
Speed in knots	26·618	24·031	15·594	12·206
Draught, mean	9·364	9·54	9·448	9·62
Displacement, tons	1023·9	1 053	1 050	1065·5
App. slip per cent.	24·605	13·67	8·25	7·355
Engines in operation.	Main turbines	Main turbines	Main and cruising	Main and cruising
S.H.P.	17 251	7 124	1 587 turbines 804-I.H.P. cruising engine	688 turbines + 423-I.H.P. cruising engine
Revs. per min.	597·06	418·63	258·31	200·29

T.S.S. T.B.D. "Nicholson" (see *Shipbuilding and Shipping Record*, 12th August 1915). 300 b.p. 305 o.a. × 30 ft. 7 in. × 30 ft. 4 in. w.l. × 9 ft. 5 in. mean draught. 1 050 tons. i.m.a. = 196·6. Area w.l. plane = 6 050. w.s. = 9 760. Tons per inch = 14·39. ω = ·426. $\otimes$ coefficient = ·684. Prismatic = ·624. l.w.l. coefficient = ·66. Cramp-Zoelly turbines. Propellers, D = 7 ft. 8½ in. Three blades. P = 6 ft. 8 in. $\frac{P}{D}$ = ·865. Projected area = 28·21. Expanded area = 31·5. Disc area = 46·67. Four forced draft Keith fans. Four White Forster boilers with eleven burners in each. 12 knots = 195 revolutions; 15½ knots = 252 revolutions; 24 knots = 415·4 revolutions; 29 knots = 563·4 revolutions.

STANDARDISATION TRIALS.

Knots.	Lbs. oil per nautical mile.	Revolutions per minute.	S.H.P.	$\frac{V}{\sqrt{L}}$.
14	140	228	1 000	
16	160	250	1 500	
18	195	292	2 400	
20	245	328	3 400	
22	310	367	4 850	
24	390	412	7 100	
26	...	...	10 400	
28				
30	...	...	...	1·732

U.S. torpedo-boat destroyer "Cummings." Twin screw. 300 ft. × 30·25 × 9·25 ft. draught. Δ = 1 010 tons. Block coefficient = ·421. Parsons turbines (with compound reciprocating engines for cruising speeds used in conjunction with cruising turbines).

Noted from Jane's *Fighting Ships*, 1914 :—

Trial.	Four hours at full speed.	Four hours' consumption trial.	Four hours at cruising speed with the reciprocating engine in use.	Four hours at 12 knots under similar conditions.
Steam pressure in lbs.	251·1	193·2	205·9	115·6
Mean revolutions per minute	615·79	420·95	256·07	196·31
Shaft H.P.	18 295	7 246	1 961	931
Mean speed in knots	30·57	23·99	15·47	11·95
Mean apparent slip per cent.	24·5	13·35	7·94	6·95
Boilers in use	4	4	1	1
Burners in use	48	19	6	4
Lbs. of oil per hour	18 284	7 965	2 354	1 340
Evaporation per lb. of oil	13·16	13·82	14·24	14·78
Lbs. oil per S.H.P. hour	·999	1·099	1·199	1·44
Lbs. water per S.H.P. hour	13·15	15·19	17·09	21·28
$\frac{\Delta^{\frac{2}{3}} V^3}{\text{Power}}$	157	192	190	184

Diameters of rotors:—H.P. = 48 in. L.P. = 60 in. Cruising turbine = 46 in. Astern turbines = 44 in. Reciprocating engine, $\frac{16 \text{ in.} - 24 \text{ in.}}{18 \text{ in.}}$. Boilers of the Normand type, 12 burners to each boiler. Main condenser cooling surface = 10 800 sq. ft. Parsons vacuum augmenter with cooling surface = 253 sq. ft.

"Alsatian." Four screws (see *Engineering*, 26th December 1913). 600 ft. l.w.l. 570 ft. b.p. × 72 × 28·5. Δ = 22 500.

600-mile trial run from Corsewall Point to the Longships and back, at 19·96 knots mean speed for the run south, tidal influences practically equally balanced, and 19·05 northward against adverse currents and strong head winds and sea. Average for the 600 miles = 19½ knots, shaft horse-power averaging 20 620, at 278 revolutions. Coal consumption = 1·3 lb. per shaft horse-power hour. On the measured mile at Skelmorlie, 19th December 1913, mean speed 20 knots = at 28 ft. 6 in. mean draught, Δ = 22 500, 285 revolutions, 21 375 S.H.P. pretty evenly divided between the four shafts.

Steam trials of H.M. armoured cruiser "Cochrane" (see *Engineering*, 13th July 1906). 480 ft. × 73 ft. 6 in. × 27 ft. 13 550 tons displacement.

Nineteen water-tube boilers of the Yarrow type, and six cylindrical boilers. Engines two sets, four-cylinder triple reciprocating.

Knots.	I.H.P.	Revolutions.	Skin H.P.	$\frac{D^{\frac{2}{3}}V^3}{I.H.P.}$.
14·3	4 911	82·75	...	338
21·37	16 080	122·35	...	344
23·292	23 649	135·2	...	303

Torpedo vedette-boats for the Roumanian Government (see *Engineering*, 19th April 1907). B.p. about 96 ft. × 13 ft. × 2 ft. 9½ in. draught. 51 tons displacement. 100 ft. over all. Two sets compound engines, screws running in tunnels. Cylinders $\frac{8\frac{1}{2}\text{ in.} - 17\text{ in.}}{9\text{ in.}}$ × 185 lbs. pressure. Propellers 3 ft. 3 in. diameter. Three-bladed.

One water-tube boiler, with oil fuel. Four hours' trial. Mean speed = 18·036 5 knots. Mean I.H.P. = 622·7. Mean revolutions per minute = 554·8. $\frac{D^{\frac{2}{3}}V^3}{I.H.P.} = 129·5$.

Hydraulically propelled steam lifeboat "President Van Heel," used at the wreck of the "Berlin" at the Hook of Holland, built by John I. Thornycroft & Co. Ltd., Chiswick, in 1895. 55 ft. overall. 53 ft. l.w.l. × 13 ft. 6 in. mld. (15 ft. over sponsons) × 5 ft. 6 in. mld. depth. Extreme draught fully loaded = 3 ft. 3 in. About 3 in. trim by the stern, keel stepped. Block coefficient seems to be ·47 (see *International Marine Engineering*, December 1907).

The load, consisting of crew, four tons of coal, mast and sails, some thirty or more passengers, and tanks full of fresh water, with the propelling machinery and boiler, gave a displacement of about 30 tons.

Thornycroft boiler, 145 lbs. working pressure per sq. in. One compound surface condensing engine, driving direct a nearly horizontal centrifugal pump, the impeller of which, 30 in. in diameter, delivered the water by which the pump was fed, by a scoop-shaped inlet amidships, through four outlets in the sides of the boat, two for ahead and two for astern; cylinders = 8½ in.

and 14½ in. × 12-in. stroke. The engine had no reversing gear: valves in the discharge pipes from the centrifugal pump controlled the direction ahead or astern. Engines designed for 250 I.H.P. The mean speed over six runs on the measured mile was 9·294 knots, ¾ knot in excess of that guaranteed by the builders, which was 8½ knots in the fully-loaded condition. On trial, 140 lbs. pressure, 449 revolutions per minute, 220 I.H.P.

Taking guarantee figures,

$$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \frac{(30)^{\frac{2}{3}} \times (8{\cdot}5)^3}{250} = 23{\cdot}7. \qquad \frac{V}{\sqrt{L}} = 1{\cdot}168.$$

The results of trial give

$$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}} = 35{\cdot}2. \qquad \frac{V}{\sqrt{L}} = 1{\cdot}277.$$

Screw ferryboat "Cincinnati" (described in the *Proceedings of the American Society of Naval Architects and Marine Engineers*, 1896, in a paper by Mr F. L. Du Bosque). Dimensions of actual vessel: L.W.L. 200 × 39·208 × 11·208 ft. extreme draught. 8-in. keel. Keel, 180 ft. long. Take the dimensions as 200 × 39·208 × 10·6 ft. mean draught. Block coefficient = 0·402. Wetted surface = 7 469 sq. ft. Displacement = 953 tons. Mid area = 244 sq. ft. Mid coefficient = 0·615 4. Coefficient of water lines = 0·756. Large ratio of displacement to wetted surface.

One screw at each end. Trial with aft screw only.

Knots.	Speed in statute miles per hour.	I.H.P.	Skin H.P.	Revolutions.	Slip per cent.	Screws removed. Lb. tow-rope resistance.	E.H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	$\frac{\text{E.H.P.}}{\text{I.H.P.}}$	Wave H.P.
5·21	6	110	23·25	...	...	2 700	43·15	125	.392	19·9
6·08	7	132	35·6	...	...	3 130	58·4	165	·442	22·8
6·945	8	175	52·2	...	...	3 880	82·8	186	·474	30·6
7·81	9	250	72·6	91	19·5	4 950	118·7	185	·475	46 1
8·69	10	364	98	103	24·5	6 400	170·5	175	·469	72·5
9·55	11	520	128·5	115	27·5	...	...	163	...	...
10·41	12	720	163·7	128	28	...	...	152	...	...

The I.H.P. is varying as the fourth power of the speed at 9·84 knots.

T.S.S. 1906 (derived). Progressive trial. Dimensions: 348 × 44·1 × 16·4 ft. mean draught. Trim 6 in. by the stern. Displacement = 5 150 tons. Block coefficient = 0·716. Mid-area coefficient = 0·932. Prismatic coefficient = 0·768.

Knots.	I.H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	Revolutions.	
8	538	285	...	
10	1 020	293	69	
12	1 900	270	84	
13	2 510	260	92	
14	3 290	249	100	Highest speed on trial.
14·5	3 740	242	104	

The I.H.P. varies as the fourth power of the speed at about 12·68 knots, but there is a hollow in the curve higher up.

S.S. ——. 260 × 36·2 × 17 ft. 3 in. mean draught (trial). Δ = 3 533 tons. ω = ·783. Mid area immersed = 572 sq. ft. Mid-area coefficient = ·943. Prismatic coefficient = ·83. Calculated wetted surface = 15 600 sq. ft.

One engine $\frac{24\frac{1}{2}\text{ in.} - 50\text{ in.}}{39\text{ in.}}$ × 120 lbs. Two S.E.B. 12 ft. 6 in. diameter × 10 ft. 6 in. Four plain furnaces, 46 in. inside diameter. G.S. = 84 sq. ft. H.S. = 2 720 sq. ft. Superheater, 72 tubes; area through tubes, 56·5 sq. in. Extended surface = 800 sq. ft. Propeller, 14 ft. 0 in. diameter. 18 ft. 3 in. pitch. 58 sq. ft. expanded surface in four C.I. blades. Solid.

Revolutions.	Knots.	I.H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	Mean pressure referred to L.P.	Steam.	Receiver.	Vacuum.	Steam temperature F.	Funnel temperature F.	Skin H.P.	Propeller K.	App. slip per cent.
61·5	9·695	864	234	36·5	120	15	26	450	650	276	306	12·59
55·75	8·99	676	250	31·4	120	10	26	450	550	...	294	10·46
49·5	8·147	486	258	25·5	...	...	...	...	...	...	284	7·85

$B_m = 13·91$. $\frac{L}{B} = 7·19$.

Wetted surface by Mumford's formula = $(260 \times 17{\cdot}25 \times 1{\cdot}7) + (260 \times 36{\cdot}2 \times {\cdot}783) = 14\,990$ sq. ft. Adding 4 per cent. gives 15 600 sq. ft.

Wetted surface by Taylor's fig. 41. $C = 16{\cdot}25$. $\frac{B}{H} = 2{\cdot}04$.

$S = C\sqrt{DL} = 16{\cdot}25 \times \sqrt{3\,533 \times 260} = 15\,600$ sq. ft.

"Chicago," twin-screw. Actual ship: $315 \times 48{\cdot}25 \times 19$ ft. mean draught. Displacement = 4 543 tons. $l = 3{\cdot}15$. $l^{3\cdot5} = 55{\cdot}4$. Wetted surface calculated = 18 460 sq. ft. Propellers, four blades. Propeller diameter = 15 ft. 6 in. Pitch = 22 ft. 6 in. Pitch ratio = 1·45. Surface ratio = 0·413. Block coefficient = 0·551. Mid coefficient = 0·868. Prismatic coefficient = 0·635.

Total weight of machinery = 937 tons, including water. Fourteen boilers, 9 ft. × 9 ft. 10 in. = 190 lbs. W.P.

Knots.	I.H.P.	Skin H.P.	Revolutions.	Apparent slip per cent.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.
15·33	4 606	1 180	70·4	10·2	214
13·27	2 793	784	59·3	7·7	230
10·47	1 441	404	46·8	7·8	218
4·32	210	...	19·9	10·34	105

I.H.P. varies as the fourth power of the speed at about 14·64 knots.

Passenger steamer. Single-screw. Actual ship: $285 \times 35 \times 15{\cdot}625$ ft. mean draught. Displacement = 2 543. Prism. coefficient = 0·633. Wetted surface = 13 000. Block coefficient = 0·594. Mid-area coefficient = 0·938.

Knots.	I.H.P.	Skin H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.
6 029	161·8	59·2	253
9 964	653·5	248	283
11 272	1 190	443	290
13 959	1 928	637	263
15 158	2 808	806	231

I.H.P. varies as the fourth power of the speed at about 13·9 knots.

T.S.S. "City of Paris" (from rough figures given by Sir W. H. White). Actual ship: $525 \times 63 \times 21 \cdot 25$ ft. mean draught. (Clipper.) Corrected here to 517 (effective length) $\times 63 \times 21 \cdot 25$ mean draught in feet. Block coefficient = 0·581. Displacement = 11 550 tons. Displacement given elsewhere as 13 000 tons at 23 ft. draught. Wetted surface calculated = 38 000 sq. ft.

Knots.	I.H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	Skin H.P.
10	2 000	255	710
14	4 600	304	1 840
18	10 000	297	3 760
20	14 500	281	5 070

I.H.P. varies as $(\text{speed})^4$ at about 19·3 knots. The speed at trial was higher than 20 knots.

T.S.S. "Normannia" (afterwards "L'Aquitaine") (from Professor W. F. Durand's book, *Resistance of Ships and Screw Propulsion*). Actual dimensions: — $498 \cdot 7 \times 57 \cdot 4 \times 22 \cdot 25$ ft. mean draught. Displacement = 10 500 tons. $\omega = 0 \cdot 582$. Mid area = 1 169 sq. ft. Mid coefficient = 0·915. Prismatic coefficient = 0·636.

Engines, three-cylinder triple, two sets, $\frac{40 \text{ in.} - 67 \text{ in.} - 106 \text{ in.}}{66 \text{ in.}}$.

Propellers, three blades, diameter = 18·12 ft. Pitch = 26·74. $\frac{\text{Pitch}}{\text{Diameter}} = 1 \cdot 48$. Area = 87·6. Area ratio = 0·313. Boiler pressure = lb. sq. in.

Knots.	I.H.P.	Skin h.p.	Revolutions.	App. slip per cent.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	Indic. thrust lb.	Lb. mean pressure ref. L.P. cylinder.
20·75	16 244	5 230	92·5	15·0	264	217 000	30
18·63	9 616	3 860	80	11·7	323	148 500	20·6
14·53	4 310	1 900	60	8·2	341	88 600	...
10·12	1 570	686	40	4·2	315	48 400	...

I.H.P. varies as $(\text{speed})^4$ at about 19 knots.

T.S.S. "City of Lowell." Pleasure steamer running on Long Island Sound. (Described in the *Proceedings of the American Society of Naval Architects and Marine Engineers*, 1895, by Professor Denton.) Built by A. Cary Smith, New York. Engines by Bath Ironworks, Maine. For 600 passengers, 420 tons freight. Actual vessel:—(L.W.L.) $319{\cdot}9 \times 48 \times 12$ ft. 10 in. mean draught. Displacement = 2 445 tons. Block coefficient = 0·434. Midship area = 467 sq. ft. Mid coefficient = 0·76. Prismatic coefficient = 0·572. Wetted surface = 13 855 sq. ft. Augmented surface = 15 399 sq. ft. Air and bilge pumps on each main engine.

Cylinders, $\dfrac{26 \text{ in.} - 40 \text{ in.} - 64 \text{ in.}}{36 \text{ in.}}$. Propellers, four-bladed solid manganese bronze, polished and sharp. $\dfrac{\text{Pitch}}{\text{Diameter}} = 1{\cdot}5$. Diameter = 11·08 ft. 23 in. diameter boss. Pitch = 16·63 ft. Projected area = 33·56 sq. ft. Expanded surface = 46·86 sq. ft. Immersion, 16 in. Both screws turn same direction. Area ratio = 0·486. Slip at 111·2 revolutions = 7 per cent.

Trials. Date.	Knots.	I.H.P.	Revolutions.	Tons displacement.	App. slip per cent.	Indic. thrust lb.	$\dfrac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Mean pressure ref. L.P.
May 29	16·2	2 727	108·1	2 546	8·73	50 000	291	21·73
May 30	19·27	4 347	125·9	2 445	6·9	68 500	299	29·65

Total feed per hour, all purposes, per I.H.P. main engines = 17·5 lb. Feed water consumed by main engines alone, per I.H.P. hour = 15·16 lb. Probable percentage of total feed consumed by auxiliaries = 11·25. Water evaporated per sq. ft. heating surface per hour lb. = 19·3 at 16·2 knots. Coal per sq. ft. grate, per hour lb. = 16·7 at 16·2 knots.

Ferry-boat "Edgewater." Propeller at each end of boat. Trial with aft screw only. (*Proceedings American Society of Naval Architects and Marine Engineers*, 1902.) Actual vessel:—L.W.L. $173 \times 34 \times 9{\cdot}8$ ft. trial draught. Displacement = 687 tons. Block coefficient = 0·417. Wetted surface = 5 764 (to base) sq. ft. Propellers, one at bow, 10·03 ft. pitch; one at stern, 10·19 ft. pitch; 8·0 ft. diameter; boss 18 in. diameter. Expanded surface = 31·9 sq. ft. Projected surface = 26·4 sq. ft.

Trial with One Screw Astern Pushing. No Propeller Forward.

	Knots.	I.H.P.	Revolutions.	Mean pressure ref. L.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	Steam lb. press.	App. slip per cent.
Up	6·84	120	78·6	9·07	208	136	13·38
Down	7·01	143	80·9	10·45	188	135	13·83
Up	8·85	209·9	99·8	12·42	257	136	11·72
Down	8·77	224·5	99·7	13·22	234	133	12·41
Up	10·23	358·5	120·7	17·45	233	133	15·7
Down	10·42	390·7	122·7	18·8	226	131·5	15·37
Down	10·72	468·5	130·7	21·15	206	137·5	18·31
Up	10·39	408·1	125·7	18·97	214	136	17·9
Down	11·5	654·5	143	26·61	181	127	19·92
Up	11·27	570	138·6	24·1	195	136·5	19·12
Up	12·52	1 015	166·5	35·2	151	123	25·1
Down	12·61	949	164	34·07	165	111	23·4

Engine cylinders, $\frac{22 \text{ in.} - 30 \text{ in.} - 30. \text{ in.}}{24 \text{ in.}}$. Piston rods, 4¼ in. diameter.

Summary.

Knots.	I.H.P. Total.	I.H.P. Skin.
6·92	131·5	40
8·81	217·2	79·2
10·28	374·6	122·8
10·55	438·3	132
11·38	612·2	163
12·56	982	215

I.H.P. varies as (speed)^4 at 11·5 knots.

2 000-ton T.S.Y. Actual dimensions:—250 × 34·45 × 14·7 ft. mean draught. Displacement = 2 000 tons. Block coefficient = 0·554. Fine midship section.

PROGRESSIVE TRIAL.

Knots.	I.H.P.	Skin H.P.	Revolutions.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	Percentage Engine Efficiency.
16·1	3 730	808	158·5	178	87·8
15·86	3 400	776	155	186	87·5
15·26	2 797	692	145·0	203	86·3
14·87	2 470	639	140·4	212	85·6
14·01	1 864	545	128·8	235	84·3
12·9	1 324	438	115·5	258	81·9
11·9	979	342	105·8	273	79·5
10·9	...	...	95·3	...	76·8
9·92	560	205	85·5	276	73·8

I.H.P. varies as $(\text{speed})^4$ approximately about 14·55 knots.

T.S.S. "Guardian" (from Professor Durand's book, *Resistance of Ships, etc.*). Actual ship:—104·5 × 20 × 7·75 ft. mean draught. Displacement = 222 tons. Block coefficient = 0·480. Mid-area coefficient = 0·748. Mid area = 116 sq. ft. Prismatic coefficient = 0·642. Wetted surface calculated = 2 378. Propellers, four blades. Pitch ratio = 1·50. Surface ratio = 0·566.

Knots.	I.H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	Skin H.P.	Revolutions.	Slip per cent.
12·33	1 060	64·8	86·3	138·6	18·0
11·84	804	75·6	76·9	128·7	15·3
9·94	374	96·1	46·8	102·8	11·0

I.H.P. varies as $(\text{speed})^4$ at about 10·55 knots.

Steam yacht, twin-screw (about 100 ft. long). 100-ft. model:—$100 \times 21 \times 7{\cdot}06$ ft. mean draught. Displacement = 184 tons. Block coefficient = 0·435. Midship area = 99·4 sq. ft. Mid-area coefficient = 0·67. Wetted surface calculated = 2 110. Prismatic coefficient = 0·65.

Knots.	i.h.p.	Skin h.p.	App. slip per cent.	$\frac{D^{\frac{2}{3}}V^3}{I.H.P.}$
5·97	42·7	9·86	20·89	161·3
7·71	82·6	20·4	20·22	180·3
8·54	118	27·13	22·7	167·2
9·06	146	32·1	23·48	165·7
9·72	200	39·2	25·2	148·8
10·29	263	46	27·3	133·7

i.h.p. varies as $(\text{speed})^4$ at about 9 knots.

North Sea trawler (see *The Shipbuilder*, December 1913). Length b.p., 92 ft. Breadth mld., 21 ft. 8 in. Depth mld., 10 ft. 8 in. Displacement = 260 metric tons. Draught forward, 7 ft. 3. in. Draught aft, 11 ft. 2 in. Block coefficient = ·486. (English tons displacement = 256.)

The displacement of 260 metric tons are made up as follows :—

Hull and equipment	136 tons.
Machinery	52 „
Bunker coal	50 „
Feed water	8 „
Ice	10 „
Drinking water	1·5 „
Crew and effects	2·5 „
	260 tons.

Engines triple, steam reciprocating, $\frac{10\text{ in.} - 16\text{ in.} - 26\text{ in.}}{17\frac{1}{2}\text{ in.}}$ 120 revolutions per minute. 230 I.H.P. Estimated speed, 9 knots. Propeller, 8 ft. diameter.

$$\frac{D^{\frac{2}{3}}V^3}{I.H.P.} = \frac{(256)^{\frac{2}{3}} \times (9)^3}{230} = \frac{38{\cdot}3 \times 729}{230} = 124\tfrac{1}{2}.$$

The midship section coefficient is frequently about ·825 in this class of vessel.

In this vessel $\frac{\text{Beam}}{\text{Mean draught}} = 2{\cdot}45$. $\frac{D}{\left(\frac{L}{100}\right)^3} = 330$. $\frac{V}{\sqrt{L}} = {\cdot}939$.

Taking $\frac{\text{Block coefficient}}{\text{Mid-area coefficient}} = \frac{{\cdot}486}{{\cdot}825}$. Prismatic coefficient = ·59.

French torpedo-boat destroyers "Fourché" and "Faulk" (see *The Shipbuilding and Shipping Record*, 6th November 1913). Length w.l., 246 ft. Breadth, 24 ft. 9 in. Length b.p., 237½ ft. Astern draught, 9 ft. 6 in. Displacement on full load, 850 tons. Draught amidships, 8 ft. 8 in. Midship section coefficient = ·760. Mean prismatic coefficient = ·768. Block coefficient = ·584. Turbines, direct-driven twin screws. Propellers, diameter = 6 ft. 11 in. Pitch, 6 ft. 5 in.

(1) Fourché. Full-power trial, six hours' duration. Displacement = 725 tons on draught. Astern, 8 ft. 3 in. Probably draught amidships = 7 ft. 5 in. Block coefficient = ·581. Mean prismatic coefficient = ·765. 680 revolutions per minute. Mean speed, 33·20 knots. B.H.P. = 18 500. Liquid fuel, 185 lbs. pressure at burners. Du Temple boilers. 10·40 tons fuel per hour. Smooth sea. Knots per ton of fuel burnt = 3·19. $\frac{B}{H} = 3{\cdot}34$. $\frac{D}{\left(\frac{L}{100}\right)^3} = 54{\cdot}1$. $\frac{V}{\sqrt{L}} = 2{\cdot}155$.

Propellers

$$K = \frac{D^2 \times \left(\frac{PR}{101{\cdot}33}\right)^3}{\text{S.H.P.}} = 418.$$

(2) 14-knot consumption trial, six hours' duration. Displacement = 725 tons. Liquid fuel, 141 lb. pressure at burners. Mean revolutions = 242 per minute. Mean speed = 14·3 knots. Knots per ton of fuel burnt = 15·38.

S.S. ——. 418·2 × 54·4 × 26 ft. draught. Block coefficient = ·755. Carrying 17 000 bales of cotton. Built in 1906.

Engines, $\frac{24\frac{1}{2}\text{ in.} - 35\text{ in.} - 51\text{ in} - 74\text{ in.}}{51\text{ in.}} \times 220$ lbs. Three S.E.B. 9 c.f. G.S., 159. H.S., 7 290. F.D., 2 250 I.H.P. usually at sea. 10⅛ knots. 62 revolutions. 30 tons coal per day (moderately good coal). 14 expansions. 12 700 tons displacement.

The engines sometimes develop 2 500 I.H.P. for ¼ knot more, *i.e.* for 10⅜ knots.

$$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \frac{544{\cdot}3 \times (10\frac{1}{8})^3}{2\,250} = 252.$$

The engines of the later ships of the line work with $16\frac{1}{2}$ expansions with greater economy and less wear and tear. Perhaps ω = ·752 is about the best commercial block coefficient for these vessels, which are $456 \times 56 \times 38$, with engines of 54-in. stroke. The draught may be increased to 29 ft. 5 in.

U.S. scout "Salem." Trials.

	Mean speed in knots.	Mean revolutions per minute.	App. mean slip per cent.	I.H.P.*	B.H.P.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Lbs. coal per I.H.P. hour.
Full speed, 4 hours	25·947	378·39	19·8	21 333	19 200	197·5	1·81
24 hours at $22\frac{1}{2}$ knots	22·536	312·535	15·7	10 378	9 340	266·5	1·78
24 hours at 12 knots	11·937	164·11	15·15	1 511	1 360	271·7	2·68

From the standardisation runs the propulsive efficiency was as follows :—

Knots.	$\frac{\text{E.H.P.}}{\text{B.H.P.}}$
12	·548
14	·564
16	·578
18	·591
20	·609
22	·62
24	·64
26	·592

The Argentine torpedo-boat destroyer "Jujuy" (see *The Shipbuilder*, December 1912). Length overall, 289 ft. 2 in. Length w.l., 286 ft. 6 in. Length b.p., 280 ft. Breadth extreme, 27 ft. Depth, 17 ft. $0\frac{3}{4}$ in. Draught normal and at trial, 8 ft. $8\frac{1}{2}$ in. Displacement normal, about 995 tons. Displacement maximum, about 1 290 tons.

Taking breadth on water-line at 26 ft. 3 in., block coefficient = ·544. Two propellers, each four-bladed ; diameter = 7 ft. 6 in., shaft centres, 10 ft. 6 in. apart. Curtis Germannia turbines, total S.H.P. = 24 000 at 640 revolutions. Contract speed, 32 knots.

Sister ships realised 34 knots average speed on six hours' trial, the power attained being in excess of the above figure.

* Equivalent I.H.P. based on assumption of 10 per cent. engine friction.

Table XXXVI.

Figures derived from a table in a paper by Mr M'Kechnie of Barrow, giving particulars of shelter-deck cargo steamers, 100 A1 at Lloyd's, showing fuel economy of large capacity ships, assuming 1·5 lb. good South Wales coal per I.H.P. hour. All at 13 knots speed.

No.	Length.	Breadth.	Depth.	Mean draught.	Displacement.	Deadweight.	Block coefficient.	Midship-area coefficient.	Prismatic coefficient.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	I.H.P.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	$\frac{V}{\sqrt{L}}$.	Tons coal per 24 hours.	Lbs. coal per 100 miles per ton deadweight.
1	390	45′ 9″	29′ 6″	24′ 6½″	8 640	5 000	·69	·974	·709	11·72	8 54	3 475	266	·659	55·9	8·0
2	415	48 9	31 0	25 6	10 240	6 000	·696	·972	·716	11·74	8·52	3 725	277	·638	59·9	7·1
3	438	51 5	32 8	26 3½	11 870	7 000	·702	·974	·721	11·73	8·53	3 970	287	·622	63·8	6·5
4	458	53 9	34 0	27 0½	13 500	8 000	·71	·972	·73	11·74	8·52	4 225	295	·608	67·9	6·05
5	475	55 9	35 5	27 11	15 100	9 000	·715	·973	·735	11·73	8·53	4 475	300	·597	71·9	5·7
6	493	58 0	36 7	28 7	16 750	10 000	·72	·971	741	11·78	8·50	4 725	305	·586	76	5·42
7	521	61 2	38 9	30 0	19 850	12 000	·728	·970	·75	11·74	8·52	5 200	311	·57	83·5	4·97
8	535	62 9	39 9	30 7	21 470	13 000	·732	·971	·754	11·72	8·54	5 430	313	·563	87·1	4·8
9	548	64 1	40 9	31 3	23 070	14 000	·736	·971	·758	11·70	8·55	5 675	314	·556	91·1	4·66
10	570	66 9	42 4	32 4½	26 150	16 000	·742	·971	·764	11·71	8·54	6 130	316	·545	98·6	4·4

More modern steamers have greater beam as percentage of length than the above examples, and the draught does not increase quite at the same rate as in this table. Restricting the draught makes propulsion more difficult, and tends to slightly modify the advantage of the large steamer. Since this paper was written, methodical model experiments with the broader ships have been made, and their greater all-round economy established.

S.S. "P." Trial. Actual dimensions: 226 × 34·16 × 12·33 ft. draught. Displacement = 1 820 tons. Calculated wetted surface = 9 980 sq. ft. Immersed mid area = 390 sq. ft. Block coefficient = 0·67. Prismatic coefficient = 0·734. Mid-area coefficient = 0·926. Propeller pitch = 14·75 ft.

Knots.	I.H.P.	Skin H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.
5·4	120	33·8	195
7·0	220	70·9	232
8·6	400	127	237
9·25	510	156	232
9·84	650	185·7	219
10·0	680	194·7	219
11·61	1 275	294	183
12·0	1 490	323	173

I.H.P. varies as $(\text{speed})^4$ at 10·52 knots.

U.S.S. "Yorktown." (Paper by Mr D. W. Taylor, American Society of Naval Architects.) Tank trials with model 20 ft. long. Displacement 2 405 lbs. in fresh water, corresponding to displacement of ship in salt water of 1 680 tons. Ship, 230 × 36 × 14 ft. draught. Block coefficient = 0·508. Resistance curves are given at various draughts of water and trim. Mid-area coefficient = 0·868. Prismatic coefficient = 0·585. Actual model:— 20 × 3·15 × 1·219 ft. mean draught, at normal draught and trim. $\frac{\text{Beam}}{\text{Draught}} = \frac{36}{14}$.

Knots.	Resistance in lbs.		
	Total.	Skin.	Residuary.
3	6·8	5·6	1·2
4	13·26	9·75	3·51
4·5	20	12·17	7·83
5	26·2	15·1	11·1
5·4	35·8	17·6	18·2
6	70	21·45	48·05

100-ft. model: $100 \times 15{\cdot}67 \times 6{\cdot}09$ ft. mean draught, normal draught and trim. Displacement = 138·3. $\frac{\text{Beam}}{\text{Draught}} = \frac{36}{14}$. Salt water.

$\frac{V}{\sqrt{L}}$.	Knots.	e.h.p.	Resistance in lbs.			Lbs. residuary resistance per ton of displacement.	©.
			Total.	Skin.	Residuary.		
·670 5	6·705	16·1	782	628	154	1·147	·856
·895	8·95	41·6	1 517	1 066	451	3·35	·93
1·007	10·07	72·1	2 333	1 326	1 007	7·5	1·139
1·119	11·19	104·5	3 042	1 614	1 428	10·6	1·205
1·208	12·08	155	4 185	1 845	2 340	17·4	1·419
1·34	13·4	346·6	8 430	2 240	6 190	46	2·31

$$\frac{\text{(Salt) Residuary resistance of 100-ft. model}}{\text{(Fresh) Residuary resistance of 12-ft. model}} = \left(\frac{100}{20}\right)^3 \times \frac{36}{35}.$$

Skin resistance of 100-ft. model = $\cdot 009\,70 \times$ wetted surface $\times V^{1\cdot 83}$. E.H.P. = total resistance $\times V \times \cdot 003\,070\,7$.

In this analysis the skin resistance of the 20-ft. model was calculated from the coefficients $f = \cdot 008\,34$ and $n = 1{\cdot}94$ given in Table I. At the Washington tank $f = \cdot 009\,7$ and $n = 1{\cdot}854$ are used, and give the same result at the highest speed (6 knots), but 3 per cent. higher values of the skin resistance at 4 knots, and 5½ per cent. higher than our values at 3 knots. In other analyses with 20-ft. models we have kept to Taylor's constants, $f = \cdot 009\,70$ and $n = 1{\cdot}854$ (Table IV).

U.S.S. "Yorktown." 100-ft. model. $100 \times 15{\cdot}67 \times 6{\cdot}09$. $\Delta = 138{\cdot}3$. $\frac{B}{H} = \frac{36}{14}$. Wetted surface = about 2 000.

100-ft. model compared with 20-ft. model. V = speed in knots.

$$\frac{36}{35} \times \frac{\text{Wave resistce. 100-ft. model}}{\text{Wave resistce. 20-ft. model}} = \left(\frac{100}{20}\right)^3 = \left(\frac{5}{1}\right)^3 = (5)^3 = 125 \times \frac{36}{35} = 128\tfrac{1}{2}.$$

TABLE XXXVII.—LIST OF 100-FT. MODELS FOR WHICH THE E.H.P. CURVE IS GIVEN.

Name of vessel.	Dis-place-ment.	Length.	Beam.	Mean draught.	Coefficients of fineness.			Approx. wetted surface.		Highest speed on our curve.
					Block.	Mid area.	Pris-matic.			
*S.S. Merkara . . .	85·3	100	10·35	4·51	·642	...	...	1 440		6·86
Rota's model No. 3 . .	121·6	100	15·3	5·56	·50	...	...	1 708		12·47
*Gunboat Argus . .	61·11	100	12·3	4·00	·439	...	...	1 220		12·4
Sir A. Denny's model A .	108·2	100	11·82	6·12	·524 2	·923 8	·567 3	1 689		12·57
,, ,, B .	85·3	100	11·82	5·10	·495 5	·908 8	·545 1	1 460		12·57
,, ,, C .	63·8	100	11·82	4·08	·463 6	·886 8	·522 7	1 243		12·57
,, ,, D .	45·85	100	11·82	3·175	·427 5	·852 7	·501 2	1 049		12·57
Popper's boat A . . .	70	100	15·62	3·48	·451	...	...	1 420		17·5
,, ,, B . . .	52·8	100	15·25	2·47	·49	...	...	1 266		14·55
,, ,, C . . .	45·8	100	15·1	2·47	·429	...	...	1 327		15·57
*Torpedo-boat Biddle . .	43·4	100	10·35	3·06	·479	·724	·663	1 030		23·98
Rota's model No. 5 . .	34·6	100	10·95	2·595	·43	...	...	911		20·8

* For the steamers marked thus, compare their E.H.P. with their I.H.P. from steam trials.

Tank trials of fine models. (From Sir A. Denny's paper to the International Engineering Congress, Chicago, 1893.)

ACTUAL MODELS.

Model.	In feet.			Lbs. displacement.	Mid area.	Wetted skin.	Coefficients.		
	Length	Moulded breadth.	Moulded draught.				Prism.	Mid area.	Block.
A	11·951	1·414 6	·731 7	405	·956 5	23·96	·567 3	·923 8	·524 2
B	11·951	1·414 6	·609 7	318·75	·784	20·83	·545 1	·908 8	·495 5
C	11·951	1·414 6	·487 8	238·6	·612 1	17·75	·522 7	·886 8	·463 6
D	11·951	1·414 6	·379 2	171	·457 5	14·96	·501 2	·852 7	·427 5

Feet per min.	Knots.	Lbs. resistance.			
		A.	B.	C.	D.
240	2·37	1·48	1·29	1·07	·94
300	2·962	2·39	2·0	1·7	1·45
340	3·357	3·18	2·75	2·22	1·81
360	3·558	3·83	3·1	2·5	2·04
380	3·75	...	3·52	2·9	2·35
400	3·95	...	4·24	3·5	2·85
420	4·147	...	5·37	4·45	3·52
440	4·346	...	7·0	5·64	4·37

Knots.	Resistance in lbs. of tank model "D."		
	Total.	Skin.	Residuary.
2·37	·94	·72	·22
2·962	1·45	1·118	·332
3·357	1·81	1·414	·396
3·558	2·04	1·58	·46
3·75	2·35	1·76	·59
3·95	2·85	1·94	·91
4·147	3·52	2·144	1·376
4·346	4·37	2·336	2·034

TABLE XXXVIII.—LIST OF VESSELS FOR WHICH THE SLOPE OF THE I.H.P. SPEED CURVE HAS BEEN MEASURED.

The point at which the I.H.P. is varying as the fourth power of the speed being termed the limiting economical speed.

Name of ship.	Displacement.	Length.	Beam.	Mean draught at trial.	Coefficients of fineness. Block.	Mid area.	Prismatic.	Wetted surface.	Limiting economical speed.	Trial speed.
(S.S.) Coasting steamer	1 370	218	32·8	9·72	·69	·95	·727	...	9·52	10·1
T.S.S. 1906	5 150	348	44·1	16·4	·716	·932	·768	...	12·68	14
S.S. Merkara	3 890	360	37·2	16·25	·642	...	...	18 660	13	12·91
Ferryboat Cincinnati †	953	200	39·208	10·6	·402	·615 4	·655	7 469	9·84	10·41
*S.S. P——	1 820	226	34·16	12·33	·67	·926	·724	...	10·52	12
*T.S.S. K——	7 270	440	48·85	16·15	·734	·915	·801	...	12½	14
*S.S. M——	1 910	270	31·3	13·92	·573	·846	·676	...	11·9	12
LL. T.S.S.	11 810	460	57·5	23	·68	·93	·731	...	15·78	16·54
(Single) Hammonia III.	5 910	373	45	20·25	·609	·927	·657	...	14·6	15·24
(Twin) Battleship Bayern	7 370	321·5	60	19·62	·682	·882	·778	...	13·45	14·29
*(Single) Inter. channel steamer	2 110	265	35·2	13·5	·586	...	...	...	13	14
(Twin) Battleship Chicago	4 543	315	48·25	19	·551	·868	·635	...	14·64	15·33
*(Single) Passenger steamer	2 543	285	35	15·625	·571	·938	·610	...	13·9	15·158
(Twin) Battleship Maine	12 500	388	72·2	23·5	·65	...	...	34 490	16·35	18·15
Edgewater (aft screw)	687	173	34	9·8	·417	...	...	‡5 764	11·05	12·56
T.S.S. Paris	11 550	517	63	21·25	·581	...	...	...	19·3	...
T.S.S. Normannia	10 500	498·7	57·4	22·25	·582	·915	·636	...	19·0	20·75

Abbreviations.—Single screw = (S.S.), or (single), or (aft screw). Twin screw = (T.S.) or (T.S.S.). Turbine, 3 screw.

* The steamers marked thus (*) were tried at a mean draught considerably less than their load draught.

† With aft screw only. ‡ To base.

TABLE XXXVIII.—LIST OF VESSELS FOR WHICH THE SLOPE OF THE I.H.P. SPEED CURVE HAS BEEN MEASURED—*continued*.

The point at which the I.H.P. is varying as the fourth power of the speed being termed the limiting economical speed.

Name of ship.	Displacement.	Length.	Beam.	Mean draught at trial.	Coefficients of fineness.			Wetted surface.	Limiting economical speed.	Trial speed.
					Block.	Mid area.	Prismatic.			
(T.S.) 184-ton Yacht . . .	184	100	21	7·06	·435	·67	·65	...	9·0	10·29
(T.S.) Ironclad Lepanto . .	14 740	400·5	72·75	30·125	·59	·896	·66	36 325	18·0	19·0
(T.S.) —— 2 000 tons . .	2 000	250	34·45	14·7	·554	Fine	...	...	14·55	16·1
(T.S.) Cruiser Argonaut . .	11 000	435	69	25·25	·51	...	...	...	19·5	21·17
(T.S.) City of Lowell . . .	2 445	319·9	48	12·81	·434	·76	·572	13 855	...	10·8
(T.S.) Cruiser Terrible . .	14 200	500	71·5	27	·515	...	...	...	21·5	22·41
(T.S.) Cruiser Good Hope . .	14 100	500	71	26·1	·533	...	...	...	21·5	23·05
(Single) Gunboat Ceram . .	510	152	25·6	8·95	·513	·783	·654	4 600	11·96	12·19
(T.S.) Cruiser Edgar . . .	7 390	360	60	23·75	·504	...	...	...	19·0	...
(T.S.) Cruiser Hermes . .	5 600	350	54	20·5	·506	...	...	...	18·7	20·5
(T.S.) Cruiser Colorado . .	13 670	502	69·5	23·92	·581	·972	·599	44 250	...	22·24
(T.S.) Despatch vessel Iris . .	3 290	300	46·08	18·08	·461	·889	·52	...	17·75	18·573
(T.S.) Cruiser Monmouth . .	9 800	440	66	24·5	·484	...	...	33 300	21·5	22·8
(T.S.) Guardian	222	104·5	20	7·75	·480	·748	·642	...	10·55	12·33
(T.S.) Cruiser Terpsichore . .	3 330	300	43	16·18	·558	...	...	...	18·33	...
(Single) Gunboat Argus . .	406	188	23	7·5	·439	...	...	...	15·1	16
(Single) Dutch tugboat . .	69	72	14·75	5·605	·406	...	...	...	9·33	11·01

Abbreviations.—Single screw = (S.S.), or (single), or (aft screw). Twin screw = (T.S.) or (T.S.S.). Turbine, 3 screw.

* The steamers marked thus (*) were tried at a mean draught considerably less than their load draught.

TABLE XXXVIII.—LIST OF VESSELS FOR WHICH THE SLOPE OF THE I.H.P. SPEED CURVE HAS BEEN MEASURED—*continued.*

The point at which the I.H.P. is varying as the fourth power of the speed being termed the limiting economical speed.

Name of ship.	Dis-place-ment.	Length.	Beam.	Mean draught at trial.	Coefficients of fineness.			Wetted sur-face.	Limiting econo-mical speed	Trial speed.
					Block.	Mid area.	Pris-matic.			
(T.S.) U.S.S. Yorktown . .	1 680	230	36·0	14	·513	·867	·591	...	16·3	16·65
(T.S.) Cruiser Barham . .	1 780	280	35	12·875	·494	...	...	...	18·65	19·53
(Turbine, 3 screw) Cruiser Amethyst	3 000	360	40	14·5	·504	...	...	...	21·33	23·4
(T.S.) Cruisers Pyramus and Pegasus	2 130	300	36·5	13·43	·506	...	...	...	19·5	21·0
(T.S.) Cruiser Medusa . .	2 800	265	41	16·5	·547	...	...	...	18·3	19·25
(Single) U.S.S. Manning . .	1 000·7	188	32·81	12·33	·46	...	...	7 273	15·1	16·0
(T.S.) British Scouts . . .	2 850	370	38·75	14·18	·491	...	...	...	22·8	25·25
(T.S.) Gunboats, Sharpshooter class	735	230	27	8·25	·503	...	...	...	18·66	20
(T.S.) Torpedo-boat Makrelen .	105	140	14·25	5·26	·35	...	...	...	...	20·0
(Single) French torpedo boat .	46·1	108	11·0	4·7*	·29*	...	...	...	...	19·3
(Single) Yarrow T.B. (built 1879)	27	86	11·0	...	...	...	...	...	...	20·0
(T.S.) U.S.T.B. Biddle . .	168	157	16·25	4·81	·479	·724	·663	...	...	30·0
(Single) T.B. Söbjörnen . .	140·5	145·5	15·5	5·815	·375	...	...	2 283	...	23·4

Abbreviations.—Single screw=(S.S.), or (single), or (aft screw). Twin-screw=(T.S.) or (T.S.S.). Qy. turbine, 3 screw.
* About.

TABLE XXXIX.—LIST OF 100-FT. MODELS FOR WHICH THE SLOPE OF THE I.H.P. SPEED CURVE HAS BEEN MEASURED.

The point at which the i.h.p. is varying as the fourth power of the speed being termed the limiting economical speed.

Name of vessel.	Displacement.	Length.	Beam.	Mean draught at trial.	Coefficients of fineness.			Approx. wetted surface.	Limiting economical speed.	Trial speed.
					Block.	Mid area.	Prismatic.			
(Single) Coasting steamer	132·5	100	15·1	4·46	·69	·95	·727	1 780	6·45	6·86
T.S.S. 1906	122·1	100	12·69	4·72	·716	·932	·768	1 710	6·8	7·5
*S.S. Merkara	85·3	100	10·35	4·51	·642	...	...	1 440	6·9	6·81
*Ferryboat Cincinnati, with aft screw only	119·0	100	19·6	5·3	·402	·6154	·655	1 867	6·95	7·38
S.S. P——	158	100	15·1	5·46	·67	·926	·724	1 995	7·0	8·0
T.S.S. K——	84·7	100	11·1	3·67	·734	·915	·801	...	5·96	6·68
S.S. M——	98·1	100	11·64	5·17	·573	·846	·676	1 560	7·25	7·33
LL. T.S.S.	121·5	100	12·5	5·0	·68	·93	·731	1 700	7·36	7·71
(Single) Hammonia III.	114	100	12·06	5·42	·609	·927	·657	1 665	7·56	7·9
T.S. Battleship Bayern	223	100	18·7	6·11	·682	·882	·773	2 310	7·5	7·99
(Single) Inter. Channel steamer	113·5	100	13·29	5·38	·586	...	...	1 646	8	8·6
(T.S.) Battleship Chicago	145·6	100	15·3	6·03	·551	·868	·635	1 860	8·25	8·65
(Single) Passenger steamer	110	100	12·29	5·49	·571	·938	·610	1 617	8·25	8·99
(T.S.) Battleship Maine	215	100	18·6	5·98	·65	...	...	2 291	8·3	9·23
Edgewater (aft screw)	132·5	100	19·65	6·05	·417	...	...	1 920	8·4	9·58

For the steamers marked thus * compare E.H.P. with I.H.P. curves.

TABLE XXXIX.—LIST OF 100-FT. MODELS FOR WHICH THE SLOPE OF THE I.H.P. SPEED CURVE HAS BEEN MEASURED—*continued*.

The point at which the i.h.p. is varying as the fourth power of the speed being termed the limiting economical speed.

Name of vessel.	Displacement.	Length.	Beam.	Mean draught at trial.	Coefficients of fineness. Block.	Mid area.	Prismatic.	Approx. wetted surface.	Limiting economical speed.	Trial speed.
T.S.S. City of Paris	83·6	100	12·2	4·11	·581	...	...	1 420	8·5	...
T.S.S. Normannia	85·5	100	11·5	4·46	·582	·915	·636	1 435	8½	9·32
(T.S.) 184-ton Yacht	184	100	21	7·06	·435	·67	·65	2 110	9	10·29
*(T.S.) Ironclad Lepanto	280	100	18·17	7·53	·59	·896	·66	2 270	9	9·5
*(T.S) ——. 2 000 tons	128	100	13·8	5·88	·554	Fine	...	1 740	9·2	10·2
(T.S.) Cruiser Argonaut	188·7	100	15·87	5·8	·51	...	...	1 800	9·85	10·16
(T.S.) City of Lowell	75·9	100	15·1	4·04	·484	·761	·572	1 370	...	10·8
(T.S.) Cruiser Terrible	118·6	100	14·3	5·4	·515	...	..	1 650	9·6	10·03
(T.S.) Cruiser Good Hope	113	100	14·2	5·22	·533	...	...	1 646	9·6	10·3
*(Single) Gunboat Ceram	145·2	100	16·85	5·89	·513	·783	·654	1 990	9·7	9·89
(T.S.) Cruiser Edgar	158·5	100	16·66	6·6	·504	...	...	1 950	10	10·54
(T.S.) Cruiser Hermes	130·8	100	15·42	5·85	·506	...	...	1 796	10	10·96
*(T.S.) Cruiser Colorado	108	100	13·85	4·77	·581	·972	·599	1 757	...	9·93
(T.S.) Despatch vessel Iris	123	100	15·4	6·0	·461	·889	·52	1 765	10·25	10·72
(T.S.) Cruiser Monmouth	115	100	15·0	5·57	·484	...	...	1 720	10·25	10·87
(T.S.) Guardian	195	100	19·12	7·42	·480	·748	·642	2 180	10·8	12·07
T.S. Cruiser Terpsichore	128·3	100	14·34	5·39	·558	...	...	1 715	10·6	11·55

* For the steamers marked thus * compare their E.H.P. with their I.H.P. curves.

TABLE XXXIX.—LIST OF 100-FT. MODELS FOR WHICH THE SLOPE OF THE I.H.P. SPEED CURVE HAS BEEN MEASURED—*continued.*

The point at which the i.h.p. is varying as the fourth power of the speed being termed the limiting economical speed.

Name of vessel.	Displacement.	Length.	Beam.	Mean draught at trial.	Coefficients of fineness. Block.	Mid area.	Prismatic.	Approx. wetted surface.	Limiting economical speed.	Trial speed.
*(Single) Gunboat Argus	61·1	100	12·3	4·00	·439	...	...	1 220	11	12·4
*(Single) Dutch tugboat	185	100	20·5	7·79	·406	...	...	2 152	11	13
*(T.S.) U.S.S. Yorktown	138·3	100	15·67	6·09	·513	·867	·591	2 000	10·75	11·0
(T.S.) Cruiser Barham	81·1	100	12·5	4·6	·494	...	...	1 400	11·15	11·68
(Turbine, 3 screw) Cruiser Amethyst	64·3	100	11·1	4·03	·504	...	...	1 245	11·25	12·33
(T.S.) Cruisers Pyramus and Pegasus	78·9	100	12·17	4·48	·506	...	...	1 400	11·25	12·1
(T.S.) Cruiser Medusa	150·5	100	15·49	6·23	·547	...	...	1 900	11·25	11·81
*(Single) U.S.S. Manning	151	100	17·5	6·39	·46	...	...	2 060	11	11·68
(T.S.) British Scouts	56·25	100	10·47	3·83	·491	...	...	1 160	11·85	13·1
(T.S.) Gunboat, Sharpshooter class	60·5	100	11·74	3·59	·503	...	...	1 200	12·3	13·2
(T.S.) Torpedo-boat Makrelen	38·3	100	10·18	3·76	·35	...	...	983	...	16·91
(Single) French torpedo boat	36·6	100	10·2	4·35	·29	...	...	...	...	18·58
(Single) Yarrow T.B. (built 1879)	42·5	100	12·8	...	...	...	...	...	...	21·6
*(T.S.) U.S.T.B. Biddle	43·4	100	10·35	3·06†	·479†	·724†	·663	1 030	...	23·98
(Single) T.B. Söbjörnen	45·7	100	10·66	4·00	·375	...	...	1 080	...	19·4

For the steamers marked thus * compare their E.H.P. with their I.H.P. curves. † Coef. water plane = ·748.

FOR THE VESSELS MENTIONED ON THIS PAGE, SEE E.H.P. OR I.H.P. CURVES.

Name.		Dis-place-ment.	Length.	Breadth.	Draught.	Mid area.	Skin.	Coefficients.			Source.
								Prism.	Mid area.	Block.	
Denny's full model	M	505	100	20	11·0	213·9	3 672	·826	·972	·803	Sir A. Denny's paper at the Chicago Congress in 1893.
,, ,, ,,	N	416	100	20	9·25	178·9	3 270	·815	·966	·787	
,, ,, ,,	O	328	100	20	7·5	144·1	2 912	·798	·960	·766	
,, ,, ,,	P	245·5	100	20	5·83	110·7	2 545	·777	·949	·738	
,, ,, ,,	Q	147·2	100	20	3·75	68·6	2 092	·751	·915	·687	
,, ,, ,,	R	57·8	100	20	1·666	27·74	1 597	·728	·833	·607	
,, 200-ft. barge		72·6	100	13·5	2·25	30·0	1 512	·846	·989	·837	Inst. N.A., 1900.
Yorktown normal model		138·3	100	15·67	6·09	82·8	about 2 000	·591	·867	·513	Taylor, Amer. N.A.
Derived from Yorktown No. 8		138·3	100	20·9	4·565	...	2 000	...	...	·513	
				(See other account of Yorktown.)							

TABLE XL.—LIST OF 100-FT. MODELS FOR WHICH THE E.H.P. CURVE IS GIVEN.

Name of vessel.	Dis-place-ment.	Length.	Beam.	Mean draught.	Coefficients of fineness. Block.	Coefficients of fineness. Mid area.	Coefficients of fineness. Pris-matic.	Approx. wetted surface.	Highest speed on our curve.
Sir A. Denny's full models, from paper to the International Engineering Congress at Chicago, 1893 — M	505	100	20	11·0	·803	·972	·826	3 672	11·4
N	416	100	20	9·25	·787	·966	·815	3 270	11·4
O	328	100	20	7·5	·766	·960	·798	2 912	11·4
P	245·5	100	20	5·83	·738	·949	·777	2 545	11·4
Q	147·2	100	20	3·75	·687	·915	·751	2 092	11·4
R	57·8	100	20	1·666	·607	·833	·728	1 597	11·4
Sir A. Denny's 200-ft. barge	72·6	100	13·5	2·25	·837	·989	·846	1 512	14·28
H.M.S. Greyhound	238	100	19·25	7·98	·534	·743	·719	2 532	9·15
*Dutch tug-boat	185	100	20·5	7·79	·406	...	...	2 152	13·0
*Ceram (gun-boat)	145·2	100	16·85	5·89	·513	·781	·656	1 990	9·75
*Ironclad Lepanto	230	100	18·17	7·53	·59	·896	·66	2 270	9·5
*U.S.S. Yorktown (*normal*)	138·3	100	15·67	6·09	·508	·868	·585	2 000	13·4
Models derived from Yorktown:									
No. 2	138·3	100	11·74	8·14	·508	(See other account with block coef. = ·527)			12·08
No. 4	138·3	100	14·36	6·65	·508				12·08
No. 5	138·3	100	16·98	5·615	·508				12·08
No. 7	138·3	100	19·57	4·87	·508	...	...	...	12·08
No. 8	138·3	100	20·9	4·57	·508	...	...	...	12·08
*Ferryboat Cincinnati	119·0	100	19·6	5·3	·402	·615 4	·655	...	7·38
*U.S.S. Manning	151	100	17·5	6·39	·48	...	...	2 060	11 68
*U.S. Cruiser Colorado	108	100	13·85	4·77	·581	·972	·599	1 757	9·93

* For the steamers marked thus, compare their E.H.P. with their I.H.P. from steam trials.

No. 2 model derived from Yorktown. Actual model:—20 × 2·35 × 1·629 ft. draught. $\frac{\text{Beam}}{\text{Draught}} = \frac{27}{18 \cdot 7}$.

Knots.	Lbs. resistance.		
	Total.	Skin.	Wave.
3	6·8	5·6	1·2
4	12·8	9·75	3·05
4·5	19·5	12·17	7·33
5	23·6	15·1	8·5
5·4	32·4	17·6	14·8

100-ft. model:—100 × 11·74 × 8·14 ft. draught. Displacement = 138·3.

Knots.	e.h.p.	Lbs. resistance.		
		Total.	Skin.	Wave.
6·705	16·0	778	628	150
8·95	39·73	1 447	1 066	381
10·07	69·3	2 241	1 326	915
11·19	91·9	2 675	1 614	1 061
12·08	137	3 695	1 845	1 850

(*See* Plate 22.)

No. 4 model derived from Yorktown. Actual model:—20 × 2·872 × 1·331 ft. mean draught. $\frac{\text{Beam}}{\text{Draught}} = \frac{33}{15 \cdot 3}$.

Knots.	Lbs. resistance.		
	Total.	Skin.	Wave.
3	6·65	5·6	1·05
4	12·8	9·75	3·05
4·5	19·5	12·17	7·33
5	24·9	15·1	9·8
5·4	33·8	17·6	16·2

100-ft. model :—100 × 14·36 × 6·65 ft. Displacement = 138·3.

Knots.	e.h.p.	Lbs. resistance.		
		Total.	Skin.	Wave.
6·705	15·62	759·2	628	131·2
8·95	39·8	1 447	1 066	381
10·07	69·3	2 241	1 326	915
11·19	97·4	2 839	1 614	1 225
12·08	143·6	3 871	1 845	2 026

(*See* Plate 22.)

No. 5 model derived from Yorktown. Actual model :—20 × 3·392 × 1·123 ft. mean draught. $\frac{\text{Beam}}{\text{Draught}} = \frac{39}{12·9}$.

Knots.	Lbs. resistance.		
	Total.	Skin.	Wave.
3	6·8	5·6	1·2
4	13·8	9·75	4·05
4·5	20·06	12·17	7·89
5	27·2	15·1	12·1
5·4	36·4	17·6	18·8

100-ft. model :—100 × 16·98 × 5·615 ft. draught. Displacement = 138·3.

Knots.	e.h.p.	Lbs. resistance.		
		Total.	Skin.	Wave.
6·705	15·94	775	628	150
8·95	43·2	1 572	1 066	506
10·07	71·5	2 312	1 326	986
11·19	107·5	3 127	1 614	1 513
12·08	155·5	4 195	1 845	2 350

(*See* Plate 22.)

No. 7 model derived from Yorktown. $\frac{\text{Beam}}{\text{Draught}} = \frac{45}{11 \cdot 2}$, for the 230-ft. vessel. Actual model:—$20 \times 3 \cdot 915 \times 0 \cdot 976$ ft. mean draught. Displacement (fresh water), 2 405 lbs.

$\frac{V}{\sqrt{L}}$.	Knots.	Lbs. resistance.		
		Total.	Skin.	Wave.
·670 5	3	7·28	5·6	1·68
·895	4	14·74	9·75	4·99
1·007	4·5	21·6	12·17	9·43
1·119	5	29·5	15·1	14·4
1·208	5·4	39·5	17·6	21·9

100-ft. model:—$100 \times 19 \cdot 57 \times 4 \cdot 87$ ft. mean draught. Displacement (salt water) = 138·3 tons.

No. 8 model derived from Yorktown. $\frac{\text{Beam}}{\text{Draught}} = \frac{48}{10 \cdot 5}$, for the 230-ft. vessel. Actual model:—$20 \times 4 \cdot 18 \times 0 \cdot 914$ ft. mean draught. Displacement (fresh water) = 2 405 lbs.

$\frac{V}{\sqrt{L}}$.	Knots.	Resistance in lbs.		
		Total.	Skin.	Wave.
·670 5	3	7·7	5·6	2·1
·895	4	15·9	9·75	6·15
1·007	4·5	22·6	12·17	10·43
1·119	5	31·5	15·1	16·4
1·208	5·4	42·2	17·6	24·6

100-ft. model:—$100 \times 20 \cdot 9 \times 4 \cdot 57$ ft. mean draught. Displacement = 138·3 tons.

Trials of tank models. (From curves in Sir A. Denny's paper to Chicago Congress in 1893.) Full model :—12 ft. long at various draughts. (Humps very pronounced at deep draughts.)

	Dimensions in feet.			Lbs. dis-place-ment.	Sq. ft. midship area.	Sq. ft. wetted skin.	Coefficients.		
	Length.	Beam.	Mld. draught.				Prism.	Mid area.	Block.
M	12	2·4	1·32	1 905	3·079	52·95	·826	·972	·803
N	12	2·4	1·1	1 555	2·55	47·1	·815	·966	·787
O	12	2·4	·9	1 239	2·073	41·95	·798	·960	·766
P	12	2·4	·7	928	1·595	36·65	·777	·949	·738
Q	12	2·4	·45	556	·988	30·15	·751	·915	·687
R	12	2·4	·2	218	·400	23·0	·728	·833	·607

Lbs. Resistance.

	Feet per min.	Knots.	M.	N.	O.	P.	Q.	R.
	240	2·37	6·6	5·6	4·7	4·0	2·7	2·0
	260	2·562	9·5	7·9	6·5	5·0	3·5	2·5
	280	2·762	13·5	11·3	9·0	6·4	4·4	3·0
Hump	300	2·962	16·0	13·3	10·6	8·1	5·7	3·7
Hollow	320	3·159	18·4	15·0	13·0	10·5	7·6	4·3
	340	3·357	25·0	21·5	18·5	14·0	9·5	5·1
	360	3·558	35·6	32·5	27·0	20·0	12·9	6·3
	380	3·75	53·2	45·3	36·5	27·0	16·9	7·3
	400	3·95	70	57·7	46·4	33·4	20·2	8·6
	420	4·147	...	67·4	53·5	38·5	23·6	9·6
	460	4·54	...	...	61·4	47·3	29·5	12·0

100-ft. models (deduced from Sir A. Denny's 12-ft. models). (From the curves in Sir A. Denny's paper to the International Engineering Congress at Chicago, 1893.)

FULL SHIPS.

	Length.	Mld. breadth.	Mld. draught.	Dis-place-ment.	Mid-ship area.	Wet skin.	Coefficients. Prism.	Mid. area.	Block.
M	100	20	11·0	505	213·9	3 672	·826	·972	·803
N	100	20	9·25	416	178·9	3 270	·815	·966	·787
O	100	20	7·5	328	144·1	2 912	·798	·960	·766
P	100	20	5·83	245·5	110·7	2 545	·777	·949	·738
Q	100	20	3·75	147·2	68·6	2 092	·751	·915	·687
R	100	20	1·666	57·8	27·74	1 597	·728	·833	·607

Model.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{B}{H} = \frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Knots.	2·37	2·762	2·962	3·357	3·558	3·75	3·95
				$\frac{V}{\sqrt{L}}$.	·685	·799	·857 5	·969	1·027	1·082	1·14
M	5	1·818	505								
N	5	2·18	416								
O	5	2·67	328								
P	5	3·43	245·5								
Q	5	5·333	147·2								
R	5	12	57·8								

U.S. battleship "Wyoming" (*The Shipbuilder*, 8, No. 27, 1912; see also paper by Lieut.-Commander H. L. Brinser, U.S.N., *Journal of the American Society of Naval Engineers*). 554 b.p. × 93 ft. $2\frac{5}{8}$ in × 28 ft. 6 in. mean draught, trial, designed. Displacement at above draught = 26 000 tons. Tons per inch = 88·41. Midship area = 2 620 sq. ft. Block coefficient = ·618. Propellers three-bladed, solid, bronze. Four shafts. Diameter propeller = 10 ft. Pitch = 8 ft. $2\frac{1}{4}$ in. Projected area = 41·06 sq. ft. 12 Babcock & Wilcox water-tube boilers. 215 lb. W.P. Total H.S. = 64 234 sq. ft. Grate surface = 1 428 sq. ft. Weight of one boiler complete including water = 58 tons. 12 F.D. (blowers) fans, Sturtevant Multivane centrifugal, double inlet type, with impellers $29\frac{1}{4}$ in. diameter outside, and running at

965 revs. per minute, capable of maintaining sufficient air for the maximum rate of combustion.

Knots.	Revs.	S.H.P. of all turbines.
10·29	146·9	2 968
12·74	181·0	5 611
15·05	214·5	8 814
17·50	255·6	15 884
18·95	277·8	19 978
20·89	308·8	27 805
21·45	321·3	32 126

Lancashire and Yorkshire Railway Co.'s turbine Channel steamers "Duke of Cumberland" and "Duke of Argyll," built by Messrs Denny, 1910. 330·7 × 41·1 × 13 ft. mean draught (equipped and loaded under service conditions). Bow rudder. Three shafts. Three-bladed propellers, D. = 5 ft. 10 in., P. = 5 ft. 3 in. Steam-driven dynamos for lighting. Five single-ended boilers, 16 ft. 6 in. diameter × 11 ft. 3 in. H.S. = 27 446 sq. ft. Grate surface = 754 sq. ft. 21 knots average speed on official trials.

Revolutions per minute.			Steam pressures.				Vacuum.	
H.P.	L.P.		Boiler steam.	H.P. turbine.	L.P. turbines.		Port.	Starboard.
	Port.	Starboard.			Port.	Starboard.		
504	499	504	151	133	16	17	28½	28½

"Ben-My-Chree." Lloyd's dimensions:—375·0 × 46 2. (See Mr Blackburn's paper, *Trans. Inst. I.N.A.*) Bow rudder to facilitate manœuvring. Four D.E.B. G.S. = 754. H.S. = 27 446. Three shafts. Power necessary for propelling the ship astern (at 16·6 knots) was about twice that required for going ahead at the same speed. Full speed ahead, 24½ to 25 knots. Propellers all of same dimensions:—Diameter = 7 ft. 2in. Pitch = 6 ft. 8 in.

Average for ten consecutive trips (five double runs) on Liverpool

service, from July 21st to 27th, 1908. Mean draught, 13 ft. 5 in. Displacement, 3 353. Douglas Head to Mersey Bar, 2 hours 19·3 minutes = 24·12 knots speed, 56 miles. Steam pressures:—Boiler steam, 164 lbs. Main steam pipe, 146½. H.P. receiver, 133·4 lbs.; P.L.P. receiver, 19·6; S.L.P. receiver, 19·5; vacuum port, 27½ in., starboard, 27 in.; revolutions per minute, H.P., 454; P.L.P., 459; S.L.P., 456.

The U.S. scout cruiser "Chester." Four shafts direct driven by Parsons' steam turbines. A description of ship, machinery, and preliminary acceptance trials is given in the May 1908 number of the *Journal of the American Society of Naval Engineers*, in an article by Lieut. A. F. H. Yates, U.S.N.

An account of the trials is given also in a paper by Mr Chas. P. Wetherbee in the *Transactions of the American Society of Naval Architects and Marine Engineers*. The average E.H.P. derived from model experiments was given in this paper, and the propulsive coefficient at full speed, in the opinion of the author of the paper, was about ·51 at full speed on the four hours' acceptance trial, but no torsionmeter measurements of shaft horse-power were made on any of the trials.

The following figures are from curves:—

Knots.	E.H.P. with appendages.	App. slip per cent.	Revolutions per minute.	Lbs. coal per E.H.P. hour.
12	750	17·3	257	5·4
14	1 200	17·3	286½	4·39
16	1 820	17·4	328	3·7
17	2 250	17·5	348	3·49
18	2 700	17·6	369	3·3
19	3 250	17·7	390	3·2
20	3 840	18	411	3·09
21	4 500	18·15	431	3·01
22	5 250	18·5	454	3·0
23	6 200	18·9	478	2·99
24	7 500	20	506	2·95
25	9 300	22	541	2·92
26	11 720	25	585	2·89
27	14 910	about 28¾	643	2·87

Designed for 24 knots. 420 ft. × 47 ft. 1⅛ in. extension × 16 ft. 9½ in. mean. Δ = 3 775 tons. 31·1 tons per inch. Beam on

L.W.L. = 46 ft. $11\frac{1}{2}$ in. Immersed midship area = 565 sq. ft. Area L.W.L. plane = 13 070 sq. ft. Wetted surface = 22 250 sq. ft. Block coefficient = ·39. Mid-area coefficient = ·73. Coefficient fineness L.W.L. = ·66. Mean prismatic coefficient = ·535. Four propellers, three-bladed, solid, manganese bronze. 6 ft. diameter × 6 ft. mean pitch. Projected area = 17·02 sq. ft. Expanded area = 19 sq. ft. Area ratio = ·673. Immersion: inboard, 5 ft. $9\frac{1}{2}$ in.; outboard, 4 ft. $9\frac{1}{2}$ in. Twelve Normand W.T. boilers.

Official four hours' trial, 26·522 knots. Mean draught, 16 ft. 6 in. Δ = 3 673 mean. Trim 8 in. by the stern. 55·08 lbs. coal per sq. ft. grate per hour. 13 300 E.H.P. 26 100 I.H.P.

Barge:—200 ft. long × 27 ft. broad. Resistance curve from experiments in Messrs Denny's tank, from Sir A. Denny's remarks on Major Rota's paper, *Trans. Inst. Naval Architects*, 1900. Model, 12 ft. long, in 18 ft. depth of water. Model, 12 × 1·62 × 0·27 ft. mean draught. Displacement in fresh water = 274 lbs. Block coefficient = ·837. Mid-area coefficient = ·989. Prismatic coefficient = ·846. Calculated wetted surface, W.S. = 22·78 sq. ft. Skin resistance = ·009 08 × W.S. × $V^{1\cdot 94}$.

Data.			Analysis.	
Speed in feet per min.	Knots.	Lbs. resistance.	Lbs. skin resistance.	Lbs. residuary resistance.
100	·998	·4	·206	·194
180	1·78	·9	·63	·27
250	2·47	2·0	1·198	·802
300	2·96	4·1	1·69	2·41
340	3·36	6·0	2·17	3·83
380	3·75	8·9	2·685	6·215
400	3·95	10·2	2·96	7·24
460	4·54	13·9	3·88	10·02
500	4·94	16·1	4·59	11·52

100-ft. model of above barge:—100 × 13·5 × 2·25 ft. mean draught. Displacement in salt water = 72·6 tons. Calculated wetted surface = 1 580 sq. ft. Skin resistance = ·009 70 × 1 580 $\times V^{1\cdot 83}$. $\dfrac{\text{Residuary resistance of 100-ft. barge in salt water}}{\text{Residuary resistance of 12-ft. model in fresh water}} = \left(\dfrac{100}{12}\right)^3 \times \dfrac{36}{35}$. The multiplier $\dfrac{36}{35}$ is used for passing from fresh water to salt water.

Knots.	Lbs. skin resistance.	Lbs. residuary resistance.	Lbs. total resistance.	E.H.P.	
2·88	118	115	233	2·06	E.H.P. = Total resistance in lb. × speed in knots × ·003 070 7.
5·14	306	160	466	7·37	
7·14	562	476	1 038	22·7	
8·55	779	1 430	2 209	56·3	
9·72	984	2 280	3 264	97·5	
10·85	1 206	3 690	4 896	163	
11·41	1 320	4 300	5 620	197	
13·1	1 700	5 960	7 660	308	
14·28	1 900	6 840	8 740	382	

"Bayern." Twin-screw. Actual ship:—321·5 × 60 × 19·62 ft. mean draught. Displacement = 7 370. Block coefficient = 0·682. Mid-area coefficient = 0·882. Prismatic coefficient = 0·773. Wetted surface calculated = 23 800 sq. ft. l = 3·21.

Knots.	I.H.P.	Skin H.P.	Revs.	App. slip per cent.	$\dfrac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Propellers.		
						No. of blades.	Pitch ratio.	Surf. ratio.
10·65	1 796	545	64·4	10·2	255	4	1·14	·402
13·69	4 122	1 103	76·9	3·4	235	4	1 06	·368
14·04	4 804	1 175	84	9·3	217	4	1·14	·402
14·29	5 488	1 238	91·6	11·2	201	4	1·09	·402

The I.H.P. varies as the fourth power of the speed at 13·45 knots.

100-ft. model of "Bayern":—100 × 18·7 × 6·11 ft. mean draught. Displacement = 223 tons. Wetted surface = 2 310 sq. ft.

U.S. B.S. "Maine." Twin-screw. (From paper by Assistant Naval-Constructor Powell, U.S. Navy.) Curve "B," or I.H.P. from mean of revolutions over measured mile, Delaware break-water, 16th July 1902. 23 ft. 2 in. mean draught. Actual dimensions (from Professor Peabody's book):—388 × 72·2 × 23·5 ft. mean draught. Block coefficient = 0·65. Displacement = 12 250 tons. Wetted surface = 34 490.

Knots.	I.H.P.	Skin H.P.	Revs.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
9	1 480	486	...	261
12·3	3 460	1 167	78·5	285
14·08	5 300	1 707	...	279
16	8 500	2 475	...	256
18·15	15 600	3 520	122	204

I.H.P. varies as $(\text{speed})^4$ at about 16·35 knots.

100-ft. model of "Maine." 100 × 18·6 × 6·05 ft. mean draught. Displacement = 210 tons. Wetted surface = 2 291 sq. ft.

First-class cruiser "Monmouth." Twin-screw. (*Engineering*, 75, 22nd May 1903.) Actual ship:—440 × 66 × 24·5 ft. mean draught. Displacement = 9 800 tons. Engines, triple, 22 000 I.H.P. at 140 revolutions. 250 lbs. steam. Thirty-one boilers. Trial, bad weather. Block coefficient = 0·484. Wetted surface calculated = 33 300 sq. ft. Propellers, diameter = 15·75 ft. Pitch = 20·0 ft. Expanded surface = 80 sq. ft. Area ratio = 0·41.

Knots.	I.H.P.	Skin H.P.	Revs.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
10·13	1 750	652	60·2	272
13·10	3 585	1 347	77·8	287
16·93	7 860	2 770	101·3	283
19·0	11 066	3 840	113·3	284
21·4	16 320	5 410	127·8	275
22·8	22 185	6 500	139	245

The I.H.P. is varying as the fourth power of the speed at about 21½ knots.

100-ft. model of "Monmouth." 100 × 15·0 × 5·57 ft. mean draught. Displacement = 115 tons. Wetted surface calculated = 1 720.

Triple-screw Japanese Trans-Pacific passenger liners "Chiyo Maru" and "Tenyo Maru." (Paper by Professor S. Terano and Baron C. Shiba, *Trans. Inst. Naval Architects*, 1911.) 550 b.p. × 63 mld. × 31 ft. 8 in. load draught (to Lloyd's Summer Freeboard). Load displacement = 21 660 tons. Block coefficient = ·691. Thirteen S.E. boilers = 15 ft. 9 in. diameter. Total grate surface = 981 sq. ft.

At 24 ft. 9 in. draught, 20·6 knots on trial, 20 000 S.H.P. with Denny-Johnson torsionmeter. Parsons turbines direct. On ordinary service 18½ knots with twelve boilers, about 18 500 S.H.P., and 1·05 lbs. fuel oil per S.H.P. hour. 1·52 lbs. best Takashima coal for same result. Ten boilers ordinarily used in service, sometimes using the forward six boilers with coal. 20 to 22 tons coal for 14 tons oil fuel.

An average result is 15·03 knots, 8 950 S.H.P., at 27 ft. 4¼ in. mean draught, with 129·5 lbs. fuel oil per day, 18 220 tons displacement, on the run from San Francisco to Honolulu. In the records of sea performances at various draughts, the shaft horsepowers named in the paper were taken from the trial power at the speeds tabulated, corrected for displacement, the correction employed assuming the shaft horse-power to vary as (Displacement)$^{\frac{2}{3}}$. See Plate 24.

H.M.S. "Barham." Cruiser. Actual ship dimensions:—280 × 35 × 13·25 ft. mean draught. Displacement = 1 830 tons. Wetted surface calculated = 11 130 sq. ft. Block coefficient = 0·495.

Knots.	Revs.	I.H.P.	Skin H.P.
10·138	100·1	551	224
14·266	143·5	1 701	579
17·553	177	3 242	1 047
19·512	201·2	5 008	1 414
20·069	210·4	5 870	1 528

Actual ship dimensions:—280 × 35 × 12·5 ft. mean draught. Displacement = about 1 730 tons. Wetted surface calculated = 10 770 sq. ft.

Knots.	Revs.	I.H.P.	Skin H.P.
10·078	101·2	616	212·4
14·164	143·9	1 899	552
17·837	183·5	3 683	1 060
19·585	204·4	5 410	1 380
19·491	202·2	5 280	1 365

I.H.P. varies as $(\text{speed})^4$ at about 18·65 knots in both cases.

100-ft. models of "Barham." Dimensions :—$100 \times 12{\cdot}5 \times 4{\cdot}74$ ft. mean draught. Displacement = 83·4 tons. Wetted surface = 1 420 sq. ft.

H.M.S. "Topaze" and H.M.S. "Amethyst." Cruisers. ("Topaze" with reciprocating engines.) "Amethyst" with turbines :—Propellers, diameter = 6·5 ft. 3 000 tons displacement. Three shafts, one screw on each. 250 lbs. per sq. in. boiler pressure. Actual vessel :—$360 \times 40 \times 14{\cdot}5$ ft. draught. Wetted surface calculated = about 16 110 sq. ft. Block coefficient = 0·504. Figures for progressive trial of "Amethyst" taken from curves in Mr Speakman's paper, *Trans. Inst. Engineers and Shipbuilders in Scotland* (1905–6).

Knots.	I.H.P.	Skin H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
23·4	14 000	3 400	190·3
23·0	12 300	3 240	206
22·0	9 550	2 850	238
21·0	7 800	2 490	247
20	6 500	2 172	256
19	5 400	1 880	264
18	4 500	1 611	270
17	3 750	1 370	273
16	3 160	1 160	270
14	2 200	787	259·5
12	1 460	512	246
10	850	306	245

I.H.P. varies as $(\text{speed})^4$ at 21·33 knots.

100-ft. model of "Amethyst":—$100 \times 11{\cdot}1 \times 4{\cdot}03$ ft. mean draught. Wetted surface = about 1 242 sq. ft. Displacement = 6·43 tons.

H.M. Scouts "Patrol" and "Pathfinder." Twin-screw. Actual vessel:—$370 \times 38{\cdot}75 \times 14{\cdot}18$ ft. draught. Displacement = 2 850 tons. Block coefficient = 0·491.

Engines $\dfrac{32\frac{1}{2}\text{ in.} - 51\frac{1}{2}\text{ in.} - 58\text{ in.} - 58\text{ in.}}{30\text{ in.}}$. 275 lbs. per sq. in. steam. $13\frac{1}{2}$-in. shaft. $6\frac{5}{8}$-in. bore. Two sheet brass condensers 6 ft. 3 in. diameter. 14 000 sq. ft. total cooling surface. 17 in. diameter circulating water inlet.

	"Pathfinder."		"Patrol."	
Knots	10·988	25·345	10·969	25·06
Revs.	85·4	220·2	84·7	213·5
I.H.P.	1 063	17 235	1 164	16 433
$\dfrac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	251	190	228	192

I.H.P. varies as $(\text{speed})^4$ at 22·8 knots.

100-ft. model of scouts:—$100 \times 10{\cdot}47 \times 3{\cdot}83$ ft. mean draught. Displacement = 56·25 tons.

H.M.S. "Good Hope." First-class cruiser. (*Engineering*, 7th March 1902.) Actual vessel:—$500 \times 71 \times 26{\cdot}1$ ft. mean draught. Displacement = 14 100 tons. Block coefficient = 0·533. Wetted surface calculated = 41 100 sq. ft.

Knots.	Revs.	I.H.P.	App. slip per cent.	$\dfrac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Skin H.P.
10·6	51	2 689	7·2	259	906
13·63	65·8	5 096	7·5	290	1 850
15·91	77·5	7 953	8·4	296	2 880
18·10	90	12 108	10·2	286	4 140
20·58	99·8	16 960	8·0	300	5 950
22·10	109·1	22 467	9·6	280	7 280
23·05	126·2	31 088	18·5	230	8 230

I.H.P. varies as $(\text{speed})^4$ at $21\frac{1}{2}$ knots.

100-ft. model of "Good Hope":—$100 \times 14{\cdot}2 \times 5{\cdot}22$ ft. mean draught. Displacement = 113 tons. Wetted surface calculated = 1 646.

H.M.S. "Terrible." First-class cruiser. Actual ship:—$500 \times 71{\cdot}5 \times 27$ ft. mean draught. Displacement = 14 200 tons. Block coefficient = 0·515. Wetted surface = 41 260 sq. ft. calculated. Propellers' diameter = 19·5 ft. Pitch = 24·0. Expanded surface = 92 sq. ft.

Knots.	I.H.P.	Skin H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	App. slip per cent.	Revs.	Mean press. referred to L.P.
13·434	5 073	1 783	280	11·0	63·71	17·9
20·964	18 500	6 280	292	14·4	103·45	40·9
22·41	25 648	7 620	257	...	112·26	52·1

The I.H.P. is varying as the fourth power of the speed at 21½ knots.

100-ft. model of "Terrible":—$100 \times 14{\cdot}3 \times 5{\cdot}4$ ft. mean draught. Wetted surface = 1 650. Displacement = 113·6 tons

H.M.S. "Iris," steel despatch vessel. Sharp entrance and run. (From Mr Wright's paper to the Inst. Naval Architects (1879). Third series of trials, 3rd July 1878.) Actual ship:—$300 \times 46{\cdot}08 \times 18{\cdot}08$ ft. mean draught. Displacement = 3 290 tons. Block coefficient = 0·461. Mid-area coefficient = 0·889. Prismatic coefficient = 0·52. 700 sq. ft. midship section immersed. Propellers, four-bladed modified Griffith's screw, twin, diameter = 16 ft. 3½ in. Pitch = 19 ft. 11½ in. Expanded surface = 144. Area ratio = 0·288.

Knots.	Revs.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	App. slip per cent.	I.H.P.	Skin H.P.
7·797	40·96	173	3·36	606	153·4
12·279	61·34	223·4	−1·63	1 833	541
16·564	85·38	196·8	1·5	5 108	1 265
18·573	97·189	183·7	2·97	7 714	1 747

I.H.P. varies as $(\text{speed})^4$ at about 17¾ knots.

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100-ft. model of "Iris":—$100 \times 15{\cdot}4 \times 6{\cdot}0$ ft. mean draught. Displacement = 123 tons.

H.M.S. "Terpsichore." Second-class cruiser. (Information from Seaton and Rounthwaite's Pocket Book. Rough figures only.) Actual ship:—$300 \times 43 \times 16{\cdot}18$ ft. mean draught. Displacement = 3 330 tons. Block coefficient = 0·558.

Knots.	I.H.P.	Skin H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
10	800	296	279
14	2 400	758	255
18	6 000	1 551	217
20	9 000	2 093	198

I.H.P. varies as $(\text{speed})^4$ at about 18·33 knots.

100-ft. model of "Terpsichore":—$100 \times 14{\cdot}34 \times 5{\cdot}39$ ft. mean draught. Displacement = 123·3. Block coefficient = 0·558.

H.M.S. "Edgar." First-class cruiser. (From Seaton and Rounthwaite's Pocket Book. Rough figures only.) Actual ship: $360 \times 60 \times 23{\cdot}75$ ft. mean draught. Displacement = 7 390 tons. Block coefficient = 0·504.

Knots.	I.H.P.	Skin H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
10	1 000	479	380
14	3 000	1 240	347
18	7 500	2 537	295
20	11 000	3 410	276

I.H.P. varies as $(\text{speed})^4$ at approximately 19 knots.

Engineering, 9th August 1901, gives:—

Knots.	Revs.	I.H.P.
11·89	55·9	1 690
13·45	63·1	2 464
16·51	79·3	5 102
18·6	92·8	8 401
20·49	106·2	13 101

100-ft. model of "Edgar":—100 × 16·66 × 6·6 ft. mean draught. Displacement = 158·5.

H.M.S. "Hermes." (Trials described in *Engineering*, June 1899.) Actual ship:—350 × 54 × 20·5 ft. mean draught. Displacement = 5 600 tons. Block coefficient = 0·506.

Knots.	I.H.P.	Cut off H.P. cyl. per cent.	Revs.	Lbs. engine steam.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
10·4	1 018	50	86·5	124	348
14·45	1 074	20	85	173	335
13·4	2 099	28	109	155	361
18·8	7 713	56	165·9	222½	272
20·5	10 224	71	182·7	229	265

I.H.P. varies as $(\text{speed})^4$ roughly at about 18·7 knots, but curve uncertain.

H.M.S. "Medusa." Third-class cruiser. (From rough figures given by Sir Wm. H. White, I.N.A., 1892.) Actual vessel:—265 × 41 × 16·5 ft. mean draught. Displacement = 2 800. Block coefficient = 0·547. [The "Pallas" (same class) made 19·25 knots at 7 610 I.H.P. $\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}} = 187$.]

Knots.	I.H.P.	Skin H.P.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	
10	700	...	284	Bad result due to shallow water.
14	2 100	...	259	
18	6 400	...	181	
20	10 000	...	159	

I.H.P. varies as $(\text{speed})^4$ at 18·3 knots.

Torpedo-boat "Makrelen." (Progressive trial described by Captain A. Rasmussen, Danish Navy.) Actual vessel:—140 × 14·25 × 7·33 ft. draught aft, 6·3 ft. mean draught normal.

127 tons displacement. Block coefficient = 0·354. Calculated wetted surface = 1 925 sq. ft.

TRIAL OF 105 TONS DISPLACEMENT.

Knots.	I.H.P.	Knots.	I.H.P.
20	1 200	16	500
18·7	1 000	15·1	400
18·2	900	14·2	300
17·6	800	12·7	200
17·15	700	10·25	100
16·6	600		

Torpedo-boat "Söbjörnen." (Progressive trial described by Captain A. Rasmussen, of the Danish Navy, in a paper to the Institution of Naval Architects, 1899.) Actual vessel:—145·5 × 15·5 × 5·815 ft. mean draught. Displacement = 140·5 tons. Block coefficient = 0·375. Calculated wetted surface = 2 283 sq. ft. $l = 1{\cdot}455$. $l^{3\cdot5} = 3{\cdot}7$.

TRIAL IN 20 FATHOMS.

Knots.	I.H.P.	Skin H.P.
23·4	2 200	505
22·0	1 800	422
21·2	1 600	379
20·0	1 310	322
18·5	1 000	258
17·6	800	222·4
16·6	600	189·7
15·0	400	141·5
12·7	200	88·6
10·0	100	45·2
6·0	47	10·7

The I.H.P. is varying as the 3·9 power of the speed at 23·2 knots.

Steam-tug "Pelorus." Single screw. 92 ft. b.p. × 20 ft. 6 in. mld. breadth × 7 ft. 10½ in. mean draught. Displacement = 213 tons. Draught 6 ft. 5 in. forward; 9 ft. 4 in. aft. On trial on the Firth of Clyde, six minutes on the measured mile = 10 knots speed. 124·5 revolutions per minute.

Engines $\frac{13 \text{ in.} - 21 \text{ in.} - 34\frac{1}{2} \text{ in.}}{24 \text{ in.}} \times 160$ lbs. W.P. Propeller three-bladed. Diameter = 7 ft. 6 in. Pitch = 11 ft. Expanded surface = 29 sq. ft.

One Scotch boiler 13 ft. inside diameter × 10 ft. mean length. 1 455 sq. ft. heating surface. 52·5 sq. ft. grate. 5 ft. 6 in. bars.

On the run from Bowling to Pará, 8·72 knots average speed, 6·56 tons coal per day; calling at Falmouth, Madeira, and St Vincent, 4 463 miles; 21 days 8 hours under way.

On trial: Coefficients. Block = ·50. Midship section = ·847. Prismatic = ·59.

S.S. "Vespasian." 275 ft. L.W.L. × 38·8 ft. beam. (Progressive trial on Hartley Mile, with reciprocating engines.) Mean draught ex keel = 18 ft. 10¾ in. Displacement = 4 350 tons. Propeller cast-iron solid, four blades. Diameter = 14 ft. Pitch = 16·35 feet. Expanded area = 70 sq. ft. Mean speeds are given for eight double runs.

Trial with reciprocating engines, $\frac{22\frac{1}{4} \text{ in.} - 35 \text{ in.} - 59 \text{ in.}}{42 \text{ in.}}$.

	Knots.	Revs.	I.H.P.	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}}$	App. slip per cent.	E.H.P. from tank.	Skin H.P.	Resid. H.P.	Residuary resistance lbs. per ton Δ.	$\frac{\text{E.H.P.}}{\text{I.H.P.}}$	Taylor's standard series resid. resistance lbs. per ton displac.
	7·5	50·58	383·7	294	7·9	176	143	33	·33	·459	...
	8·195	55·3	473·5	310	8·1	240	182	58	·53	·508	...
	8·684	58·85	582·2	299	8·35	289	216	64	·555	·497	...
	9·075	61·7	673	296	8·52	339	246	93	...	·504	...
	9·316	62·05	681	317	8·7	370	265	105	·845	·544	...
	9·537	64·63	769·5	302	8·9	409	284	...	...	·533	...
	9·923	67·83	903·7	290	9·3	481	316	165	1·25	·533	...
	10·204	70·05	993	286	9·74	550	328	222	1·63	·554	...
Tank	10·5	...	...	...	...	622	382	240	1·71	...	...
Tank	11·0	...	...	...	...	785	422	363	2·47	...	...
Tank	11·5	...	...	...	...	1 000	472	528	3·44	...	...

Estimated wetted surface = 16 900 sq. ft.

S.S. "Vespasian" (*continued*). Progressive trial off the Tyne, 11th April 1910, with turbines geared to the original shaft and same propeller as with reciprocating engines, viz. 14 ft. diameter × 16·35 ft. pitch. Vessel loaded to the same draught and displacement, 4 350 tons, as before.

Knots.	Revs.	S.H.P.	App. slip per cent.	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{S.H.P.}}$	Water consumption per hour, main engines.	Water consumption per hour, all purposes.	Water consumption per shaft horse-power, main engines.	E.H.P. from tank.	$\frac{\text{E.H.P.}}{\text{S.H.P.}}$
8·4	56·5	456	7·79	348	9 070	9 670	19·8	260	·57
9·56	65	740	8·88	315	12 000	12 620	16·2	415	·561
10·5	71·3	980	8·7	315	14 480	15 120	14·8	623	·636
10·66	73·3	1 095	9·74	295	15 670	16 370	14·3	667	·61

Plotting a fair curve for I.H.P. and speed from the results of the trial with reciprocating engines, and another curve for S.H.P. and speed from the trial of 11th April 1910, both with the same propeller, we obtain the following estimate of the ratio $\frac{\text{S.H.P.}}{\text{I.H.P.}}$:—

Knots.	I.H.P.	S.H.P.	$\frac{\text{S.H.P.}}{\text{I.H.P.}}$.
8	445	390	·876
8·5	535	480	·898
9	650	600	·923
9·5	765	725	·948
10	925	838	·906
10·25	1 025	905	·884

The ratio is, therefore, roughly about ·90 at 10 knots, and ·94 at 9½ knots, the usual speeds of the vessel on service.

S.S. "Vespasian" (*continued*). Progressive trial off the Tyne, 9th January 1911, with new propeller, 14 ft. diameter × 14·14 ft. pitch, four blades, 72 sq. ft. expanded blade area. Displacement, 4 350 tons, as before.

Knots.	Revs.	S.H.P.	App. slip per cent.	$\frac{\Delta^{\frac{2}{3}}V^3}{S.H.P.}$	Water consumption, main engines, lbs. per hour.	Water consumption, lbs. per S.H.P. hour.	E.H.P.	$\frac{E.H.P.}{S.H.P.}$
9·31	68·4	630	2·51	343	10 400	16·5	375	·595
9·66	71·2	720	2·82	334	11 510	15·98	429	·595
9·94	73·7	815	3·31	323	12 590	15·45	480	·59
10·34	77	945	3·54	313	14 000	14·81	580	·614

Results obtained on voyages from the Tyne to Antwerp and Rotterdam, with original propeller 16·35-ft. pitch. Displacement, 4 560 tons, say 19 ft. 9 in. mean draught, ex keel.

Knots.	Revs.	S.H.P.	App. slip per cent.	$\frac{\Delta^{\frac{2}{3}}V^3}{S.H.P.}$	Water consumption, main engines, lbs. per hour.	Water consumption, lbs. per S.H.P. hour.
9·22	64·9	736	12·03	295	12 300	18·0
9·27	63·85	710	10·0	308	11 730	16·5
9·35	65	740	10·95	304	12 140	16·4
9·37	62·9	668	7·6	338	11 100	16·6
9·61	64·8	735	8·05	333	11 890	16·2
10·22	70·6	960	10·29	308	14 510	15·1
10·58	73	1 080	10·2	301	15 680	14·5

"Monitoria." Ship with corrugated sides (see *Shipbuilder*, 4, No. 16 (1910)). 279 ft. × (40 ft. 1½ in. beam + 1 ft. 10 in. = 41 ft. 11½ in.) × 17 ft. 5 in. draught. Displacement, 4 450 tons. I.H.P. = 1 012. Revolutions, 65·8. 9·78 knots. Wetted surface, 17 480 sq. ft. Progressive trial. Mean draught, 17 ft. 10 in. Displacement = 4 575 tons.

I.H.P.	Revs.	Knots.
521	53·77	8·185
871	62·75	9·397
966	65·1	9·686
1 120	67·35	9·962
1 195	68·58	10·122

Single-screw engines, $\frac{21 \text{ in.} - 33 \text{ in.} - 56 \text{ in.}}{36 \text{ in.}} \times 180$ lbs. pressure. Two boilers, 13 ft. × 10 ft., with 3 000 sq. ft. total heating surface.

Sister ships to the "Monitoria," with plain sides (see *The Shipbuilder*, vol. iv., No. 16, 1910). Single-screw. 279 ft. × 40 ft. 1½ in. × 17 ft. 8½ in. mean draught. Displacement = 4 450 tons. I.H.P. = 1 116. 70 revolutions. 9·78 knots. Wetted surface = 17 435 sq. ft.

Machinery same as in "Monitoria."

$$\frac{\Delta^2 V^3}{\text{I.H.P.}} = 226.$$

SHALLOW WATER.

The resistance of a ship in shallow water is often enormously greater than in deep water, because of the effect of the bottom upon the streamlines, and of the wave system accompanying the vessel.

The problem has been the subject of discussion at meetings of the Institution of Naval Architects. In the discussion on Sir W. H. White's paper on "Notes on Recent Experience with some of H.M. Ships," Mr R. E. Froude mentioned experiments tried in the Admiralty tank at Torquay to determine the increase of resistance thus caused. A false bottom set at various depths in the tank caused the resistance of models to vary considerably. With the water as shallow in proportion to the size of the model as the water in Stokes Bay in proportion to a ship of from 3 000 to 5 000 tons—there was an increase of resistance of from 3 per cent. to 5 per cent., nearly constant at all speeds. Mr Froude pointed out that of the two elements of this difference in resistance, there is first the element that is nearly a constant percentage at all speeds, "attributed to the fact that the water in getting out of the way of the ship has to move in two dimensions instead of three dimensions; the motions are consequently more accentuated, and involve higher streamline speed against the ship's side, and cause greater friction." There is also the other element, viz., that due to the effect of shallow water on the wave, and which is more marked at high speeds. See also Mr D. W. Taylor's paper to the Institution of Naval Architects, spring 1894, "On Ship-shaped Stream Forms." In April 1895, Mr. D. W. Taylor read a paper to the I.N.A. "On Solid Stream Forms, and the Depth of Water necessary to avoid abnormal Resistance of Ships." Professor

Lamb, "On the Motion of Fluids," p. 117, gives a description of simple stream systems in three dimensions. Mr R. E. Froude, in his remarks at the end of Mr. Taylor's paper, pointed out that there is an increase of resistance at all speeds, even at those speeds at which there is no resistance due to wave-making. In order to properly consider the question of whether the shoal water increases those features of the streamline disturbance which are the cause of wave-making resistance (and this is at certain speeds only), the relation between the depth of water and the length of the wave that is proper to the speed of the ship has to be considered.

In a valuable paper to the Institution of Naval Architects in 1899 on "Some Steam Trials of Danish Ships," Captain A. Rasmussen, of the Royal Danish Navy, gave progressive speed and power curves of the torpedo boats "Söbjörnen" and "Makrelen" at four different depths of water. The "Söbjörnen" was 145 ft. 6 in. long × 15 ft. 6 in. beam, and 140 tons displacement, with a draught of water 3 ft. 10 in. forward, and 7 ft. 9½ in. aft. Engines: four-cylinder triple, 220 lbs. working pressure. At normal draught the displacement was 132 tons. At half power the loss in speed in shallow water was very great, while at full power the speed was higher for depths below or above 8 fathoms, this being, of the four named depths, the most disadvantageous for the propulsion of this boat at full power. It was pointed out that the speed corresponding to the points of inflexion in the curves was practically the speed v of "the wave of translation as given by the formula

$$v = \sqrt{gh}$$

where h = depth of water
and g = acceleration of gravity."

While the wave of translation in shallow water at half power is unusually high and long, it vanishes completely at full speed.

At normal draught (132 tons displacement) "Söbjörnen":—

Depth of water.	Knots at 2 200 I.H.P.	Knots at 1 000 I.H.P.
2½ fathoms . . .	24·1	13·1
6½ fathoms . . .	23·8	17·2
8 fathoms . . .	22·8	18·3
20 fathoms . . .	23·6	18·6

Söbjörnen.

Speed in knots .	6	10	12	14	16	18	20	22	24·1
I.H.P. in 20 fathoms	45	98	160	286	520	840	1 320	1 960	..
I.H.P. in 8 fathoms .	45	98	164	310	570	..	1 408	1 785	..
I.H.P. in 2¼ fathoms	..	100	260	1 080	1 180	1 220	1 380	1 660	2 200

The dotted curve on Plate 28 shows the E.H.P. for a trial in shallow water of Popper's boat A (see p. 252). When passing suddenly into shallow water, the speed of a steamer may suddenly jump from, say, 12¼ to 14½ knots (see Curve), the power remaining constant.

"The Resistance of some Merchant-ship Types in Shallow Water," paper by Professor Herbert C. Sadler, read at the American Society of Naval Architects and Marine Engineers, 16th November 1911.

The models were tested in water of varying depth in the tank at the University of Michigan. Both full and fine types were tested, including some broader types, and one with V-shaped sections. The results were given in curves representing residuary resistance in pounds per ton of displacement. Professor Sadler found that the speed at which maximum resistance occurred was a function of depth of water rather than size of ship. The first hump in the curve for a given depth of water occurred at nearly the same speed for all types. A set of curves was given for a full type of cargo boat. With the V-shaped section the hump was not so pronounced as in the types with fuller midship sections, perhaps because the mean draught of the midship section was less. The hump for a given depth of water occurred at slightly higher speeds in the fuller forms.

In *Zeitschrift des Vereines Deutscher Ingénieure*, 10th December 1904, will be found a report of an important paper entitled "Experiments to ascertain the Influence of the Depth of Water on the Speed of Torpedo Boats," by Ship-Constructor Paulus, read at the Schleswig-Holstein District Club.

The paper referred to Captain Rasmussen's papers to the Institution of Naval Architects, 1894 and 1899, and to later towing experiments with models and with full-sized ships at various depths of water to ascertain the model resistances, by Major Rota, ship constructor of the Italian Navy, and by Ship-Constructor Schütte at Bremerhaven. Rota and Schütte varied the depths of water in their tanks by constructing a movable bottom of smooth wooden planks. This bottom separated completely the

upper part of the basin from the lower, so that the particles of water could not escape below.

Rota's model-resistance curves resembled the I.H.P. curves quite remarkably, Paulus pointing out that a still better comparison would have been obtained if the corresponding E.H.P. curves were put alongside the I.H.P. curves of torpedo-boat S 119.

In Schütte's paper, under the heading "Torpedo Boat" with "A," the E.H.P. appeared to be equal at speeds 1·95 m. and 2·5 m., while the intermediate values were smaller.

Normand's formula was used for calculating the wetted surfaces of these torpedo boats.

Paulus used the terms "effective efficiency," "indicated efficiency," and "actual efficiency," meaning E.H.P., I.H.P., and $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ respectively.

PAULUS WITH TORPEDO-BOAT "S 119."

SPEEDS IN KNOTS AT EQUAL POWERS.

Depth of water.	60 m.	40 m.	25 m.	15 m.	10 m.	7 m.
5 680 I.H.P. .	27·17	26·93	26·55	27·2	27·66	27·82
4 600 ,, .	25·13	24·86	24·56	23·58	23·5	25·95
4 000 ,, .	24·01	23·73	23·46	21·80	23·52	24·52
2 700 ,, .	21·48	21·28	21·09	20·00	17·75	15·84
2 000 ,, .	20·00	19·84	19·72	19·08	16·94	15·11
1 500 ,, .	13·75	18 67	18·58	18·20	16 68	14·84
1 000 ,, .	17·16	17·16	17·06	16·86	16·22	14·44
500 ,, .	12·2	12·2	12·2	12·2	12·2	12·2

I.H.P. AT EQUAL SPEEDS.

Depth of water.	60 m.	40 m.	25 m.	15 m.	10 m.	7 m.
27 knots . .	5 590	5 715	5 920	5 605	5 315	5 195
24 ,, . .	3 995	4 140	4 290	4 710	4 110	3 870
21 ,, . .	2 465	2 560	2 650	3 550	3 525	3 510
18 ,, . .	1 240	1 250	1 285	1 410	2 815	3 210
15 ,, . .	600	600	600	615	640	1 800
12 ,, . .	285	285	285	285	285	285

SPEEDS IN KNOTS AT USUAL NUMBERS OF REVOLUTIONS.

Depth of water.	60 m.	40 m.	25 m.	15 m.	10 m.	7 m.
270 revs. per min.	26·75	26·55	26·22	26·57	27·15	27·31
250 ,, ,,	24·84	24·66	24·34	23·60	25·03	25·32
200 ,, ,,	20·52	20·36	20·42	19·56	17·55	16·05
150 ,, ,,	16·66	16·65	16·65	16·52	16·14	14·55
100 ,, ,,	11·7	11 7	11·7	11·7	11·7	11·7

Mr A. F. Yarrow, in *Cassier's Magazine*, November 1908, mentioned that the depth of water in feet to be avoided was approximately represented by the expression $\frac{(\text{speed in knots})^2}{10}$, and showed a curve for critical combinations of speed and depth of water, ordinates depth, and abscissæ speed at which the length of the transverse waves (which travel at the same speed as the ship) became indefinite. Mr Yarrow mentioned that the diverging waves, which have a speed less than the speed of the ship, did not come under the above heading, but that at very high speeds the effect of these should not be neglected.

E.H.P. curves. From deep-water progressive trials of three boats, A, B, C. (Paper by Popper, *Trans. Inst. Naval Architects*, 1905.) See forms given on Table XXXVII.

BOAT A.

Actual dimensions:—73·47 × 11·48 × 2·558 ft. mean draught. Block coefficient = 0·451. Displacement = 27·75 tons. Wetted surface = 766·39 sq. ft. Deep water figures only noted.

Knots.	E.H.P.	Skin H.P.	Wave H.P.
6	5	...	...
7	8	...	...
8	12·5	..	...
9	18	...	...
10	30	...	...
11	46	20·27	25·73
12	82	26·2	55·8
13	118	32·4	85·6
14	157	40·2	116·8
15	190	48·7	141·3

100-ft. model of boat A:—100 × 15·62 × 3·48 ft. mean draught. Displacement = 70. Block coefficient = 0·451. Wetted surface = 1 420 sq. ft.

Knots.	e.h.p.	Skin h.p.	Wave h.p.
7·0	14·7	...	...
8·17	23·4	...	...
9·34	36·6	...	...
10 5	52·5	33	...
11·68	87·8	44·1	...
12·84	134·1	58·5	75·6
14·0	238·8	74·8	164
15·2	345·3	93·3	252
16·35	458·6	115·4	343·2
17·5	553·8	138·8	415

Boat B (Popper).

Actual dimensions :—92·98 × 14·17 × 2·296 ft. mean draught. Displacement = 42·3 tons. Block coefficient = 0·49. Wetted surface = 1 087 sq. ft.

	Dia.	Pitch.	Exp. surf.
1st propeller, 3 blades .	2·788	3·214	3·293
2nd propeller, 3 blades .	2·788	3·608	3·293

2nd very much better than 1st one. Maximum efficiency, 58·2 per cent. at about 10 knots.

Knots.	E.H.P.	Skin H.P.	Wave H.P.
5	4·5	...	...
7	11	...	...
8	15	...	...
9	25	...	...
10	37½	...	...
11	56	...	...
12	82	...	...
13	117	...	...
14	168	...	...

100-ft. model:—100 × 15·25 × 2·47 ft. mean draught. Wetted surface = 1 266. Block coefficient = 0·49. Displacement = 52·8 tons.

Knots.	e.h.p.
5·19	5·84
7·26	14·3
8·3	19·45
9·34	32·3
10·38	48·4
11·41	72·3
12·47	105·9
13·5	151·2
14·55	217

BOAT C (Popper).

Deep-water progressive trials. Tow-rope resistance curve, and E.H.P. curve. (*Trans. Inst. Naval Arch.*, 1905.)

Actual boat:—92·98 × 14·04 × 2·296 ft. mean draught. Displacement = 36·6 tons. Wetted surface = 1 140 sq. ft. Block coefficient = 0·429.

Knots.	E.H.P.	Knots.	E.H.P.
6	7	11	46
7	11	12	68
8	15	13	98
9	24	14	137
10	35	15	177

100-ft. model:—100 × 15·1 × 2·47 ft. mean draught. Displacement = 45·8 tons. Wetted surface = 1 327 sq. ft. Block coefficient = 0·429.

100-ft. Models: — Deduced from results from forms on Table XXXVII.

Model.	In feet. Length.	Moulded breadth.	Moulded draught.	Tons displacement.	Mid area.	Wetted skin.	Coefficients. Prism.	Mid area.	Block
A	100	11·82	6·12	108·2	66·9	1 680	·567 3	·923 8	·524 2
B	100	11·82	5·10	85·3	54·85	1 460	·545 1	·908 8	·495 5
C	100	11·82	4·08	63·8	..	1 243	·522 7	·886 8	·463 6
D	100	11·82	3·175	45·85	..	1 049	·501 2	·852·7	·427 5

Model B.

Knots.	e.h.p.	Resistance in lbs. Total.	Skin.	Residuary.	Residuary resistance in lbs. per ton of displacement.	Ⓒ
6·85	13·71	652	478·6	173·5	2·037	·939
8·575	25·95	985·5	720	265·5	3·116	·904
9 71	41	1 377	907	470	5·51	·987
10·29	49·05	1 554	1 010	544	6·38	·996
10·85	58·4	1 754	1 111	643	7·54	1·006
11·4	72·7	2 080	...	...	...	...
12·0	102	2 767	1 331	1 436	16·85	1·299
12·57	143	3 708	1 448	2 260	26·5	1·586

Model C.

Knots.	e.h.p.	Resistance in lbs. Total.	Skin.	Residuary.	Residuary resistance in lbs. per ton of displacement.	Ⓒ
6·85	11·31	538·3	408·3	130	2·04	·94
8·575	22·08	838	614	224	3·52	·937
9·71	32·7	1 098·5	773	325·5	5·1	·955
10·29	39·1	1 237	861	376	5·89	·961
10·85	47·7	1 435	946	489	7·66	·999
11·4	61·5	1 756	1 036	720	11·29	1·11
12·0	84·2	2 282	1 135	1 147	18	1·301
12·57	114	2 955	1 234	1 721	2·7	1·539

(*See* Plate 22.)

100-ft. model (deduced from Sir A. Denny's fine model D; tank trial). (See paper read before the International Engineering Congress at Chicago, 1893.)

Model D.

Knots.	e.h.p.	Resistance in lbs.			Lbs. residuary resistance per ton of displacement.	C
		Total.	Skin.	Residuary.		
6·85	10	476·6	343·6	133	2·905	1·04
8·575	18·89	718	518	200	4·37	1·00
9·71	26·52	890·5	652	238·5	5·21	·965
10·29	31·76	1 004	726	277·5	6·06	·975
10·85	39·35	1 179	823	356	7·79	1·03
11·4	50·7	1 450	902	548	11·97	1·045
12·0	66·9	1 816	986	830	18·14	1·292
12·57	88·5	2 300	1 072	1 228	26·8	1·49

(*See* Plate 22.)

Dutch opium-cruiser "Argus." Progressive trials. (*Transactions Inst. Engineers and Shipbuilders in Scotland*, 1893, paper by Dr Robert Caird on "Propeller Diagrams.") Actual ship:— 188 × 23·0 × 7·5 ft. mean trial draught. Displacement = 406 tons. Block coefficient = 0·439. See Dr Caird's curves for slip, wake factor, propeller efficiency, engine efficiency, hull efficiency, revolutions, I.H.P., E.H.P., thrust deduction, etc. One propeller, diameter = 7·5 ft. Pitch = 9·25 ft. 205 revolutions. Designed for 16 knots at 1 024 I.H.P. Wake factor = 0·26. E.H.P. = 640. Propulsive efficiency = 0·595.

Knots.	I.H.P.	Revs.	Lbs. indicated. thrust.	E.H.P.	Skin H.P.	Wave H.P.
6	62·5	70·5	3 165	25	19·8	5·2
8	117·5	95·0	4 420	60	45	15
10	207·5	120	6 170	116	84·6	31·4
12	350	146	8 560	208	140·8	67·2
14	605	173	12 480	362	217	145
16	1 024	205	17 860	634	320	314
17	1 375	222	...	850	378	472

I.H.P. varies as $(\text{speed})^4$ at 15·1 knots.

100-ft. model of "Argus":—100 × 12·3 × 4·00 ft. mean draught. Displacement = 61·11 tons. Block coefficient = 0·439.

Percentage of top speed.	Knots.	Revs.	Percentage engine efficiency.	Percentage of full power.
37·5	4·38	96·8	53·5	6·1
50	5·84	130·2	63	11·46
62·5	7·3	164·5	71	20·3
75	8·76	200	77	34·3
87·5	10·21	237	82	59·2
100	11·67	281	85·3	100
...	12·4	304	86·5	...

(*See* Plate 35.)

Dutch tugboat. (From Mr W. F. Durand's book, *Resistance and Propulsion of Ships*, 1911, or *The Steamship*, Oct. 1897.) Values of E.H.P. determined from model experiments. Actual ship:—dimensions, 72 × 14·75 × 3 ft. 10 in. draught forward, 7 ft. 4½ in. draught aft., 5 ft. 7¼ in. draught mean. Displacement = 69 tons. Block coefficient = 0·406. Propeller pitch = 7·63 ft. Wetted surface = about 1 117 sq. ft. $\frac{\text{T.H.P.}}{\text{I.H.P.}}$ varies with increase of speed from 0·64 to 0·69. $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ varies from 0·543 to 0·462.

Knots.	I.H.P.	T.H.P.	E.H.P.	Skin H.P.	Wave H.P.	App. slip %.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	$\frac{\text{E.H.P.}}{\text{I.H.P.}}$
6·97	31·03	19·76	15·8	...	...	...	184	·509
8·07	50·56	33·16	27·42	...	...	...	174	·543
9·02	80·24	53·22	42·74	...	...	...	154	·533
10·07	132·35	89·43	70·69	...	...	...	131	·534
10·47	170·83	118·85	87·75	...	...	...	114	·514
10·84	230·58	161·4	108·46	...	..	...	93·4	·471
11·01	260·32	180·2	120·22	...	...	...	86·3	·462

I.H.P. varies as (speed)^4 at about 9·33 knots.

100-ft. model of Dutch tugboat:—100 × 20·5 × 7·79 ft. mean draught. Displacement = 185 tons. ω = 0·406. Wetted surface = 2 152 approximate.

Knots.	e.h.p.	Skin h.p.	Wave h.p.
8·22	48·85	24·8	24·05
9·51	85·1	37·9	47·2
10·65	140·6	52	88·6
11·89	220·8	70·2	150·6
12·37	274·4	78·4	196
12·8	340·1	87·1	253
13·0	377·2	91·2	286

Towing trials of "Greyhound," described by Mr Wm. Froude, *Transactions Inst. Naval Architects,* 1874, H.M.S. "Active" (3 078 tons, 4 055 horse-power, 15 knots measured mile speed) towed H.M.S. "Greyhound" (1 157 tons displacement), at nearly 13 knots speed, from the end of a boom 45 ft. long, without any difficulty in steering.

Particulars of "Greyhound":—

	Mean draught.	Midship area.	Tons displacement.	Square feet immersed skin.
	ft. in.			
Normal displacement, tons	13 9	339	1 161	7 540
Medium displacement, tons	12 11½	313	1 050	7 260
Light displacement, tons .	12 1	284	938	6 940

At normal displacement, 13 ft. 9 in. draught (No. 2), 1 161 tons.

Feet per min.	Resistance in lbs. Without bilge keels.	Resistance in lbs. With bilge keels.
1 200	19 080	20 000
1 100	14 270	14 300
1 000	10 277	10 000
900	7 320	7 030
800	5 464	
600	3 050	
500	2 140	

The bilge keels were 100 feet long and 3 ft. 6 in. wide. The extra resistance when these were fitted was less than that caused by the skin friction alone by about a half at 10 knots.

At the lighter draughts the entrance and run, of course, became finer.

"Greyhound." Single-screw sloop. Towed through still water from a long outrigged boom. (For towing trials, see *Trans. Inst. Naval Architects*, **15** [1874], and Thearle's *Theoretical Naval Architecture*, p. 347.) Actual vessel :—172·5 × 33·18 × 13·75 ft. mean draught. Wetted surface = 7 540 sq. ft. Displacement = 1 200 tons. Block coefficient = ·534. $\dfrac{\Delta}{\left(\dfrac{L}{100}\right)^3} = 238$. $\dfrac{\text{Beam}}{\text{Draught}} = 2{\cdot}41$. Prismatic coefficient =

Knots.	$\dfrac{V}{\sqrt{L}}$.	Ⓒ	Tons tow-rope resistance.	Resistance in lbs. Total.	Skin.	Wave.	Residuary resistance in lbs. per ton of displacement.
4	·302	·98	·6	1 344	890	454	
6	·454	·101	1·4	3 138	1 890	1 248	
8	·605	·101	2·5	5 606	3 230	2 376	
10	·755	·122	4·7	10 520	4 850	5 670	
12	·906	·162	9·0	20 140	6 730	13 410	

E.H.P. varies as (speed)^4 at about 10·83 knots.

Tank trials were also made with models in fresh water, and the resistances plotted in a curve.

The calm-air resistance of the "Greyhound" at 10 knots, without masts and rigging, was found by Wm. Froude to be about one and a half per cent. of the total water resistance. In merchant, passenger, and cargo vessels to-day, where there is a much greater above-water area exposed to wind, the air resistance is of course a much larger item.

100 ft.-model of "Greyhound" :—100 × 19·25 × 7·98 ft. mean draught. Wetted surface = 2 532. Displacement = 238 tons. Block coefficient = 0·534.

Knots.	e.h.p.	Residuary h.p.	Lbs. resistance. Total.	Skin.	Wave.
3·05	...	...	...	...	83·5
4·57	9·0	3·42	640 3	397	243·3
6·1	21·2	8·65	1 133	670	463
7·63	49·6	25·9	2 119	1 013	1 106
9·15	112·8	73·4	4 019	1 404	2 615

e.h.p. varies as (speed)^4 at about 8½ knots (V_m).

Tank trials were also made with models in fresh water, and the resistances plotted in a curve. The resistance curve for the full-sized ship, deduced from the tank-trial results, and corrected for friction, represents her resistance in fresh water. After being corrected for salt water, the results agreed with those obtained by towing the actual ship through smooth salt water.

(*See* Plate 22.)

Italian ironclad "Lepanto." (*Trans. Inst. Naval Architects*, 1888.) Actual ship :—400·5 × 72·75 × 30 125 ft. mean draught at trials.* Displacement = 14 740 tons. Wetted surface = 36 325 sq. ft. Twin screws, Admiralty type. Diameter = 20·5. Three blades. Pitch = 20·5. Pitch ratio = 1. Flat surface = 80 sq. ft.

[At 28·33 ft. mean normal draught, midship area = 1 843 sq. ft. Displacement = 13 851 tons. Block coefficient = 0·588. Prismatic coefficient = 0·659. Mid-area coefficient = 0·894.]

* *At trial draught the results are:*

The speeds at which the I.H.P. seems to vary as (speed)^4 are approximately 14·2 knots and 18 knots. [In the tables the extreme breadth is given, viz. 74 ft.]

Knots.	I.H.P.	E.H.P.	Skin H.P.	Wave H.P.	Tons net resistance.	Revs.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	$\frac{\text{E.H.P.}}{\text{I.H.P.}}$	$\frac{\text{Speed}}{\sqrt{l}}$.
6	700	210	160	50	5·1	32	210	·30	3
7	1 000	...	...	...	...	...	...	...	...
9	1 810	670	512	158	11·3	47·5	241	·37	4·5
10	2 450	...	...	...	...	...	...	...	...
12	4 060	1 680	1 147	533	20·7	62	251	·414	6
15	8 540	3 700	2 140	1 560	34·4	77	240	·433	7·5
16	10 300	...	...	...	...	...	...	...	...
18	14 600	6 900	3 604	3 296	57·7	91·3	241	·473	9
18·45	16 100	...	...	...	...	...	...	...	...
19	19 300	9 740	4 160	5 580	75	96·6	235	·505	9·5

100-ft. model of "Lepanto":—100 × 18·17 (18·5 extreme) × 7·53 ft. mean draught. Displacement = 230 tons. Take block coefficient = 0·59. Wetted surface = 2 270 sq. ft. [(At ext. draught) Block coefficient = 0·578. (At mld. draught) Block coefficient = 0·588. Take prismatic coefficient = 0·66; take mid-area coefficient = 0·896—these coefficients do not agree with above.]

Knots.	e.h.p.	Skin h.p.	Wave h.p.	Lbs. net resistance.
3	...	...	·391	178·5
4·5	6·018	4·76	1·235	395·5
6	14·96	10·78	4·17	725
7·5	32·16	20·0	12·2	1 205 . . Hump
9	59·3	33·6	25·76	2 020
9·5	82·9	39·7	43·6	2 625

i.h.p. might vary as $(\text{speed})^4$ at two or three different parts of the curve. For instance, at 9 knots (V_m) rising up out of a hollow, or at 7·1 knots mounting a hump.

(*See* Plate 22.)

Models tested at the experimental tank of the Royal Dockyard of Spezzia, Italy. Towed at various speeds to determine the influence of depth of water on the resistance of the ship. (From a paper by Major Giuseppe Rota, read before the Institution of Naval Architects, 1900.) No. 3 model, at full depth of water. Actual dimensions in feet:—12·24 × 1·87 × 0·68 ft. mean draught. Displacement = 488·3 lbs. Block coefficient = 0·50.

Speed in knots.	Resistance in lbs.		
	Total.	Skin.	Wave.
4·352	7·72	4·01	3·71
4·28	7·05	3·88	3·17
4·08	5·51	3·53	1·98
3·889	4·74	3·2	1·54
3·5	3·526	2·635	·891
3·11	2·713	2·048	·665
2·721	1·871	1·573	·298

No. 3, 100-ft. model :—100 × 15·3 × 5·56 ft. mean draught. Displacement = 121·6. ω = 0·50.

Knots.	e.h.p.	Residuary e.h.p.	Lbs. resistance.	Skin resistance.	Wave resistance.
12·47	141·5	78	3 687	1 667	2 020
12·25	126	65·1	3 347	1 617	1 730
11·68	91·6	38·6	2 561	1 482	1 079
11·12	75·1	28·6	2 200	1 360	840
10·01	49·4	14·9	1 606	1 120	486
8·9	34·5	9·9	1 263	900	363
7·79	20·86	3 88	871·4	709	162·4

(*See* Plate 22.)

Model No. 5. Torpedo boat. Tested in tank at Spezzia, Italy. (See *Trans. Inst. Naval Architects*, paper by Major Giuseppe Rota.) Actual dimensions :—12·33 × 1·35 × 0·32 ft. draught. Displacement = 145·2 lbs. Block coefficient = 0·43. l = 0·123 3. $\sqrt{l}$ = 0 351. l^3 = 0·001 872.

Knots.	Resistance. in lbs.	Skin resistance.	Wave resistance.
7·2	11·02	5·9	5·12
6·8	9 83	5·13	4·7
6·32	8·7	4·48	4·22
5·83	7·5	3·83	3·67
5·345	6·05	3·2	2·85
4·86	4·74	2·66	2·08
4·38	3·304	2·2	1·104
3·89	2·424	1·733	·691
3·4	1·763	1·353	·410
2·916	1·278	·985	·293
2·43	·883	·694	·189

100-ft. model of above :—100 × 10·95 × 2·595 ft. mean draught. Displacement = 34·6 tons. Block coefficient = 0·43.

Knots.	Resistance in lbs.			e.h.p.	Wave h.p.
	Total.	Skin.	Wave.		
20·8	6 212	3 480	2 732	396	174
19·37	5 560	3 050	2 510	331	149·4
18·0	4 924	2 672	2 252	272	124·2
16·6	4 253	2 293	1 960	217	100
15·23	3 482	1 962	1 520	162·5	70·9
13·86	2 766	1 656	1 110	117·8	47·1
12·48	1 948	1 359	589	74·5	22·5
11·1	1 589	1 220	369	54·2	12·6
9·69	1 079	860	219	32·1	6·51
8·3	801·4	645	156·4	20·4	3·98
6·93	574	473	101	12·2	2·15

See curve for humps and hollows.

TABLE XLI.—WAKE FRACTION w (adapted from Mr Luke's Curves).

Block coef.	Single screw.	Twin screws.				
		Bossing sloped 0° to horizontal.	Bossing sloped 22° to horizontal.	Model without bossing.	Bossing sloped 45° to horizontal.	Bossing sloped 67½° to horizontal.
·35	·064	·058	·035	·02	·013	
·36	·069	·062	·04	·024 5	·017	
·37	·074	·067	·045	·029	·021 5	0
·38	·08	·073	·05	·034	·025 5	·005
·39	·085	·078	·055	·038	·03	·008 5
·40	·09	·083	·06	·043	·035	·012
·41	·095	·088	·064 5	·048	·039	·016
·42	·10	·093	·069	·052	·043 5	·02
·43	·105	·098	·074	·057	·047 5	·023
·44	·11	·103 5	·078	·062	·052	·027
·45	·116	·109	·083	·066	·056	·03
·46	·121	·114	·087	·070 5	·061	·035
·47	·126	·119	·092	·075 5	·065	·039
·48	·131	·124 5	·097	·08	·07	·042
·49	·136	·13	·102	·085	·074 5	·046
·50	·142	·135	·107	·09	·079	·05
·51	·147	·14	·112	·095	·083	·054 5
·52	·152	·145	·116 5	·10	·087	·058
·53	·158	·15	·121	·105	·092	·062
·54	·163	·155	·126	·11	·096	·065 5
·55	·168	·160 5	·131	·115	·10	·07
·56	·174	·166	·136	·119	·105	·074
·57	·179	·171	·141	·124	·11	·077
·58	·184	·176	·146	·129	·115	·081
·59	·19	·181	·151	·134	·119	·085
·60	·195	·186	·156	·139	·123	·089

The curves given in Mr Luke's paper to the I.N.A., 1917, represented values of w_p, the wake percentage in Mr Froude's nomenclature. The table gives values of $w = \frac{w_p}{1 + w_p}$.

TABLE XLI.—WAKE FRACTION w (adapted from Mr Luke's Curves) —*continued.*

Block coef.	Single screw.	Twin screws.				
		Bossing sloped 0° to horizontal.	Bossing sloped 22° to horizontal.	Model without bossing.	Bossing sloped 45° to horizontal.	Bossing sloped 67½° to horizontal.
·61	·20	·191	·16	·143	·127	·093
·62	·205	·197	·166	·148	·132	·096 5
·63	·21	·202	·171	·153	·136	·10
·64	·215	·207	·176	·158	·14	·105
·65	·221	·213	·181	·163	·145	·108 5
·66	·226	·218	·185	·167 5	·15	·112
·67	·231	·223	·19	·172 5	·155	·116
·68	·237	·228	·195	·177 5	·159	·12
·69	·242	·233	·20	·182	·164	·124 5
·70	·247	·238	·205	·187	·168	·128
·71	·252	·244	·21	·192	·172	·131 5
·72	·257	·249	·215	·196	·176	·135 5
·73	·263	·254	·22	·201	·181	·14
·74	·268	·259	·225	·205	·186	·143 5
·75	·274	·265	·23	·211	·19	·147
·76	·279	·27	·234 5	·215	·195	·151
·77	·284	·275	·239	·22	·199	·155
·78	·29	·28	·244	·225	·203	·159
·79	·295	·286	·249	·23	·208	·164
·80	·30	·291	·254	·235	·212	·166 5
·81	·305	·296	·259	·24	·216	·171
·82	·31	·301	·264	·244	·221	·175
·83	·316	·306	·269	·249	·225	·179
·84	·321	·311	·274	·254	·23	·183
·85	·326	·316	·278	·259	·235	·186

The curves given in Mr W. J. Luke's paper to the I.N.A., 1917, represented values of w_p, the wake percentage in Mr Froude's nomenclature. The above table gives values of $w = \dfrac{w_p}{1+w_p}$.

Ship.	Length.	Beam.	Draught.	Δ.	Coefficients.				
					Block.	Mid area.	Prismatic.		
							Fore body.	Aft body.	Mean.
Ambrose	375·2	47·8	23·5	7 654	·636	·966	...	...	·659
Anselm	400·4	50·1	23·5	...	·68	·961	...	...	·708
Basil	338	43·7	23·5	7 495	756	·976	...	...	·775
Benedict	345	43·5	22·792	8 180	·836	·975	...	...	·859
City of Rome	542	52	21·5	11 200	·647	·911	...	...	·71
Derived destroyer	400	40	10·6	1 920	·396	·762?	·51	·54	·52 ?
Dominic	322	42·3	22·33	...	·778	·983	...	...	·791
Francis	355	49·25	23·5	9 120	·777	·98	...	...	·794
Gregory	26·5	40·0	20·25	4 622	·755	·978	...	...	·773
Hilary	418·5	52·2	23·5	9 300	·637	·956	...	...	·664
Hildebrand	440·3	54·1	23·5	10 195	·637	·973	...	...	·656
Huayna	260·35	36·25	18·0	3 391	·698	·955	...	...	·73
Justin	355	48·7	23·5	8 930	·767	·976	...	...	·785
Luke's model	400	58·8	19·6	7 650	·581	·855?	·63	·72	·68 ?
Manco	300·3	45·2	18	5 008	·72	...	...	...	...
Manning	188	32·7	12·3	1 000	·48	·797	...	...	·604
,,	188	32·7	12·3	1 000	·48	·797	...	...	·604
Mercantile vessel	400	50	19·5	7 200	·646	·98 ?	·68	·65	...
,, ,,	400	54·0	19·3	8 400	·705	·98 ?	·72	·74	·72 ?
Merchant vessel	400	70·1	27·0	12 860	·60	·98 ?	...	...	·613?
Michael	300·5	45·3	...	6 240	·78	...	...	...	...
Polycarp	340	46·5	23	8 000	·76	·975	...	...	·78
Alban	375·5	51·7	25·08	10 534	·757	·985	...	...	·769
Passenger vessel	400	57	18	6 400	·545	·98 ?	·59	·61	...
Stephen	376·5	50·3	23·5	9 932	·781	·977	...	...	·800
Vessel A	400	58·1	18·7	5 800	·467	·78	...	...	·60

FACTOR.

Wake Fraction.						No. of screws.	Knots speed.	$\frac{\text{E.H.P.}}{\text{I.H.P.}}$	Hull efficiency.	Source.
w calculated from Taylor's formula.	The same converted to Froude's nomenclature (w_p).	As given in Froude's nomenclature (w_p) from trials.	The same converted to *w*, a fraction of ship's speed as in Taylor's.	*w* calculated from Gordon's slide-rule.*	*w* from M'Dermott's formula.					
·268	·367	...	...	·217	·202	1	15	...	...	Calculated.
·29	·409	...	...	·25	·224	1	14	...	...	,,
·328	·488	...	...	·303	·239 2	1	10·5	...	...	,,
·368	·58	...	...	·366	·277	1	9·5	...	...	,,
·273	·376	·33	·248	·225	·259 6	1	18·2	·50	...	Baker.
– ·018	– ·016	– ·01	– ·010 1	...	·098 5	2	33	...	·98	,,
·339	·512	...	...	·32	·242	1	10·5	...	...	Calculated.
·338	·511	...	...	·317	·241 6	1	10·5	...	...	,,
·327	·488	...	...	·304	·223 5	1	9·5	...	...	,,
·15	·178	...	...	·084 2	·103 7	2	14·2	...	...	,,
·15	·178	...	...	·088 4	·098 8	2	14·6	...	...	Calculated.
·299	·426	...	...	·26	·213 1	1	10	...	...	,,
·333	·50	...	...	·31	·242	1	10·5	...	...	,,
·12	·137	·20	·166 7	...	·137 9	2	...	...	1·02	Baker.
·31	·45	...	...	·277	...	1	11·5	...	...	Calculated.
·186	·23	·11	·099 1	·092	·194 1	1	10	·49	·90	Baker.
·186	·23	·12	·107 1	·094 1	·194 1	1	15	·54	·90	,,
·155	·184	·15	·130 5	...	·105 2	2	...	...	·95	,,
·188	·231	·20	·166 7	...	·116	2	...	...	·99	,,
·13	·15	·20	·166 7	...	·077 6	2	15·2	..	1·04	,,
·34	·515	...	...	·324	...	1	10	...	...	Calculated.
·33	·492	...		·31	·242	1	10·7	...	...	,,
·328	·488	...	...	·278	·239 5	1	11·5	...	...	,,
·10	·111	·04	·038 45	...	·080 9	2	.	...	·98	Baker.
·34	·515	...	..	·323	·256	1	10·5	...	...	Calculated.
·188	·232	·18	·152 5	...	·238	1	19	·50	·98	Baker.

* Subject to 8.

TABLE XLII.—AMOUNT TO BE ADDED TO THE EFFICIENCY

Nominal pitch ratio.	Expanded area ratio									
	·25.	·26.	·27.	·28.	·29.	·30.	·31.	·32.	·33.	·34.
·8	·013 9	·013 6	·013 3	·013 0	·012 7	·012 2	·011 7	·011 1	·010 5	·010 0
·9	·010 3	·010 0	·009 8	·009 4	·009 1	·008 8	·008 3	·008 0	·007 5	·007 0
1·0	·007 9	·007 7	·007 5	·007 2	·007 0	·006 8	·006 5	·006 1	·005 9	·005 6
1·1	·005 5	·005 3	·005 2	·005 1	·005 0	·004 8	·004 7	·004 4	·004 3	·004 0
1·2	·003 1	·003 0	·003 0	·003 0	·002 9	·002 9	·002 8	·002 7	·002 6	·002 5
1·3	·002 2	·002 1	·002 0	·002 0	·002 0	·002 0	·001 9	·001 8	·001 8	·001 7
1·4	·001 2	·001 2	·001 15	·001 13	·001 12	·001 01	·001 00	·001 0	·001 0	·001 0
1·5	0	0	0	0	0	0	0	0	0	0

TABLE XLIII.—AMOUNT TO BE SUBTRACTED FROM THE EFFI-

Nominal pitch ratio.	Expanded area ratio								
	·46.	·47.	·48.	·49.	·50.	·51.	·52.	·53.	·54.
·8	·001 6	·003 1	·004 6	·006 2	·008 0	·009 8	·011 9	·013 9	·015 9
·9	·001 3	·002 6	·004 0	·005 2	·006 7	·008 2	·009 8	·011 5	·013 1
1·0	·001 1	·002 0	·003 0	·004 0	·005 1	·006 3	·007 5	·008 8	·010 0
1·1	·000 8	·001 5	·002 2	·003 0	·003 9	·004 8	·005 7	·006 7	·007 7
1·2	·000 5	·001 0	·001 5	·002 0	·002 7	·003 2	·003 9	·004 6	·005 3
1·3	·000 4	·000 7	·001 1	·001 4	·001 8	·002 2	·002 8	·003 3	·003 8
1·4	·000 3	·000 5	·000 7	·000 9	·001 1	·001 3	·001 7	·002 0	·002 2
1·5	0	0	0	0	0	0	0	0	0

FROM CURVE, WHEN THE AREA RATIO IS LESS THAN ·45.

in three blades.											Nominal pitch ratio.
·35.	·36.	·37.	·38	·39.	·40.	·41.	·42.	·43.	·44.	·45.	
·009 2	·008 5	·007 9	007 0	·006 0	·005 2	·004 3	·003 3	·002 2	·001 1	0	·8
·006 7	·006 1	·005 7	·005 0	·004 4	·003 9	·003 1	·002 3	·001 5	·000 8	0	·9
·005 2	·004 9	·004 4	·004 0	·003 5	·003 0	·002 4	·001 9	·001 2	·000 6	0	1·0
·003 8	·003 6	·003 2	·003 0	·002 6	·002 2	·001 7	·001 3	·000 9	·000 5	0	1·1
·002 4	·002 2	·002 1	·002 0	·001 7	·001 4	·001 0	·000 8	·000 6	·000 4	0	1·2
·001 7	·001 6	·001 6	·001 5	·001 2	·001 0	·000 8	·000 6	·000 4	·000 3	0	1·3
·000 9	·000 85	·000 8	·000 8	·000 7	·000 6	·000 5	·000 4	·000 2	·000 1	0	1·4
0	0	0	0	0	0	0	0	0	0	0	1·5

CIENCY FROM CURVE, WHEN THE AREA RATIO EXCEEDS ·45.

in three blades.										Nominal pitch ratio.
·55.	·56.	·57.	·58.	·59.	·60.	·61.	·62.	·63.	·64.	
·018 0	·020 0	·022 3	·024 6	·026 9	·029 1	·031 2	·034	·036 5	·039 5	·8
·014 9	·016 7	·018 4	·020 3	·022 2	·024 1	·026 0	·028	·030	·032 2	·9
·011 3	·012 7	·014 0	·015 4	·016 9	·018 2	·019 8	·021	·023	·024 5	1·0
·008 6	·009 8	·010 8	·011 9	·013 0	·014 2	·015 4	...	...	...	1·1
·006 0	·006 9	·007 6	·008 4	·009 1	·010 1	·011 0	·012	·012 9	·013 5	1·2
·004 3	·004 9	·005 5	·006 0	·006 6	·007 2	·007 8	...	...	...	1·3
·002 6	·003 0	·003 3	·003 6	·003 9	·004 2	·004 6	·005	·005 5	·006	1·4
0	0	0	0	0	0	0	0	0	0	1·5

The French quadruple-screw Atlantic liner "France." (See *The Shipbuilder*, 7, 11.) Lloyd's dimensions: Length b.p. 689 ft. × breadth 75·6. Load draught 29 ft. 10 in. Displacement = 26 760 tons. Four shafts, 250 revolutions per minute. Parsons turbines. Total heating surface boilers 99 200 sq. ft. 120 furnaces. Total grate surface 2 548 sq. ft. On her 24 hours' trial the vessel is said to have attained an average speed of 25 knots with about 47 000 S.H.P. (at what draught of ship we do not know). Midship section coefficient = ·972.

Diesel oil-engined ship "Annam," built 1913. Installation almost identical with "Selandia." Twin-screw. Built by Burmeister & Wain, Copenhagen. 425 ft. overall × 55 ft. beam × 38 ft. 6 in. depth. Net register tonnage 3 325. Total carrying capacity 9 400 tons on a draught of 26 ft. 4 in. Average speed $11\frac{1}{2}$ knots at sea. Double bottom carrying 1 254 tons of oil, not including peaks. In No. 4 hold a deep tank between the two tunnels is capable of storing 80 tons of oil, and is used as an emergency tank in case of accident to the double bottom. Total crew 32 men, of which there are 7 engineers, 2 electricians, 6 greasers—15 in all in the engine department. Main engine cylinders $23\frac{15}{64}$ in. × $31\frac{1}{2}$ in. 125 revolutions. H.P. pump of compressed air service driven from main engines. Two 8-cylinder Diesel oil engines, 3 200 (or 2 550 according to *Internal Combustion Engineering*), B.H.P. 126 revolutions per minute. Two 4-cylinder 300 B.H.P. Diesel auxiliary engines, each driving a D.C. dynamo, 220 volts, and a large air compressor. Electric-driven emergency compressor, ballast pump, two sanitary and bilge pumps. Pumps, steering gear, winches, and windlass are electrically driven. Full speed consumption of oil, including the two auxiliary motors, 10·8 tons. Reversing the main engines is effected in two seconds. A small 30 B.H.P. Tuxham hot-bulb engine, driving a dynamo at 110 volts, is used for lighting the vessel at night in port when the winches are not in use. When the large auxiliaries are employed, the lighting circuit passes through a transformer from 220 to 110 volts. The winches, etc., work at 220 volts. Fourteen winches and a warping winch aft. Electric windlass by Clarke Chapman. Hele-Shaw electric-hydraulic steering gear. A small vertical steam boiler, fired by oil, placed between the two thrust blocks, supplies steam to the room heaters and galley and fire extinguishers. Two electrically driven lubricating pumps and two water circulating pumps are placed at the forward end of the engine-room. Settling tank pump. Turning engine.

Mr D. W. Taylor, in a paper read before the American Society

of Naval Architects and Marine Engineers in 1910 on "The Effect of Parallel Middle Body upon Resistance," deals with a series of experiments with models at the U.S. Model Basin to determine, from the point of view of resistance, the most suitable length of parallel middle body for full vessels of low and moderate speeds. The following notes are taken from *The Shipbuilder*, vol. iv. The models tested all had a midship section coefficient of ·96, ordinary sections of shape shown by a drawing, and a ratio of beam to draught of 2·5. Three series were tried having prismatic or longitudinal coefficients of ·68, ·74, and ·80 respectively. Four sizes of models were used in each series to show the effect of different ratios of length to beam, and for each size of model five curves of sectional areas were used, corresponding to different percentages of parallel body. Altogether sixty models were tested. Skin frictional resistance, which is the major factor in the type of vessel under consideration, is only affected by a very small amount, about $1\frac{1}{4}$ per cent., by the variations of form. As regards residuary resistance, however, the experiments showed that there was a most suitable length of parallel middle body for minimum resistance, varying with the speed and prismatic coefficient, but not greatly affected by the size of model, *i.e.* the ratio of length to breadth. Curves were given showing the length of middle body at different speeds for minimum residuary resistance, and curves also showing the variation which can be made in this length without increasing the residuary resistance 10 per cent., or the total resistance 3 per cent., assuming the residuary resistance to be 30 per cent. of the total. In his paper Mr Taylor says: "While the results, strictly speaking, refer only to models similar to the parent form used, and the actual residuary resistances given are not perhaps the minimum that may be obtained, I think there is little doubt that, as regards desirable length of parallel middle body from the point of view of resistance, they should apply with reasonable approximation for almost any type of form such as would be used for full vessels. . . . Broadly speaking, from the point of view of resistance alone, for the range of speeds attained in practice by full vessels, the optimum length of parallel middle body is for a longitudinal coefficient of ·68 from 12 to 16 per cent., but it may be made 25 per cent. without material increase in resistance. For a longitudinal coefficient of ·74 the optimum length of parallel middle body is from 24 to 27 per cent., but it may be made from 36 to 40 per cent. without material increase of resistance. For a longitudinal coefficient of ·80 the optimum length of parallel middle body is from 32 to 35 per cent., but it may be made from

44 to 48 per cent. without material increase of resistance. These conclusions apply to values of speed-length coefficient above ·50. For very low-speed vessels the residuary resistance is such a small percentage of the total that the limits above may evidently be materially exceeded."

The curves on Plate 13 show the relation between speed-length ratio $\frac{V}{\sqrt{L}}$ and prismatic coefficient and block coefficient for actual service speeds. The uppermost curve represents "highest economical speeds" taken from Mr G. S. Baker's book *Ship Form, Resistance and Screw Propulsion*, 1915. The speeds at the right hand are for torpedo-boat destroyers.

$$C = \frac{\text{E.H.P.}}{\Delta^{\frac{2}{3}} V^3} \times 427 \cdot 1. \qquad \text{Let } \frac{\text{E.H.P.}}{\text{I.H.P.}} = \rho.$$

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = \frac{\rho \times 427 \cdot 1}{C}.$$

If
$$\frac{\text{E.H.P. (naked model)}}{\text{I.H.P. of ship}} = \cdot 50 = \rho,$$

we have
$$\text{I.H.P.} = \frac{\Delta^{\frac{2}{3}} V^3 \times C}{213 \cdot 5}$$

or
$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = \frac{213 \cdot 5}{C} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

If $\rho = \cdot 46$, as in many cases, then

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}} = \frac{196 \cdot 5}{C} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

f
$$\frac{\text{E.H.P.}}{\text{S.H.P.}} = \cdot 55,$$

s often the case with direct turbines in a smooth sea, then

$$\frac{\Delta^{\frac{2}{3}} V^3}{\text{S.H.P.}} = \frac{235}{C} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

r G. S. Baker's (1913) Set C, model 18*a*:—400 ft. ship (mercantile ship forms). Block coefficient = ·685. Prismatic = ·699. $)^3$ = 149·7. One of Mr Baker's economical speeds would be

$$V = 1{\cdot}34 \times \sqrt{\frac{{\cdot}699 \times 400}{2}}$$
$$= 15{\cdot}81 \text{ knots.}$$

Let us consider $\dfrac{V}{\sqrt{L}} = {\cdot}70$ or $V = 14$ knots as the trial speed

(K) = about 1·77.

(K)	V.	Percentage of trial V.	Likely values of ρ.	E.H.P.	I.H.P.	$\dfrac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
1·4	11·07	79	·438	1 115	2 550	240
1·5	11·86	84·6	·452	1 289	2 850	264
1·6	12·66	90·4	·461	1 632	3 540	259
1·7	13 44	96	·452	1 940	4 200	261
1·8	14·22	101·7	·46	2 222	4 830	269
1·9	15·02	107·3	·452	2 672	5 910	260

The values of ρ and trial speed are taken from Plate 38.

Mr G. S. Baker's (1913, mercantile ship forms) Set E, model 19*b* :—400-ft. ship. Block coefficient = ·805. Prismatic = ·824. $\dfrac{\Delta}{\left(\dfrac{L}{100}\right)^3} = 176.$ One of Mr G. S. Baker's economical speeds is

$$V = 1{\cdot}34 \times \sqrt{\frac{{\cdot}824 \times 400}{4}}$$
$$= 12{\cdot}18 \text{ knots.}$$

Let us take trial speed $V = 10{\cdot}69$, $\dfrac{V}{\sqrt{L}} = {\cdot}534$ for this form.

(K) = 1·31.

(K)	V.	Percentage of trial V.	Likely values of ρ.	E.H.P.	I.H.P.	$\dfrac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
1·1	8·92	83·5	·456	649	1 421	251
1·2	9·73	91	·463 7	855	1 849	250
1·3	10·54	98·8	·47	1 104	2 350	250
1·4	11·36	106.2	·473	1 448	3 060	241
1·5	12·17	113·9	...	1 847	3 890	234

In averaging results of trials to obtain the "mean of the means," the results are tabulated in the order in which they are run, thus:—

Measured speed.	First mean.	Second mean.	Third mean.	Fourth mean and average, or "mean of the means."
V_1				
	$\frac{V_1+V_2}{2}$			
V_2		$\frac{V_1+2V_2+V_3}{4}$		
	$\frac{V_2+V_3}{2}$		$\frac{V_1+3V_2+3V_3+V_4}{8}$	
V_3		$\frac{V_2+2V_3+V_4}{4}$		$\frac{V_1+4V_2+6V_3+4V_4+V_5}{16}$
	$\frac{V_3+V_4}{2}$		$\frac{V_2+3V_3+3V_4+V_5}{8}$	
V_4		$\frac{V_3+2V_4+V_5}{4}$		
	$\frac{V_4+V_5}{2}$			
V_5				

This has been the approved method on the Clyde and elsewhere for upwards of a generation. The mean so obtained differs slightly from the arithmetic mean $\frac{V_1+V_2+V_3+V_4+V_5}{5}$, but is more accurate.

Plotting Trial Analysis Results upon a Base of Percentages of Full Speed.

Some standard curves intended for the use of shipowners' staffs are shown on Plates 35 to 39.

The American practice of running standardisation trials is an admirable one, particularly if the trials are at load draught.

A diagram derived from one of these trials, showing mean pressure referred to L. P. cylinder (for reciprocating-engined steamships) as ordinates, plotted upon a base of percentages of full-speed revolutions per minute or percentages of ship's full speed as abscissæ, is invaluable. When no standardisation trial results are obtainable, spots can always be plotted, taken from indicator diagrams or torsionmeter readings on voyage. There will be a curve for each draught of ship, showing mean pressures, consumptions, etc., corresponding to fully loaded condition, partly loaded and lightly loaded. Such curves for a number of vessels will be found to resemble one another very closely when the abscissæ are percentages of full speed, or full power, or full-power revolutions, or full-speed revolutions per minute, and by the aid

of such diagrams the performance of a ship can be predicted with some accuracy. A diagram of this description can easily be drawn for any ship fitted with an indicator or a torsionmeter. What is necessary is for the engineer to give a correct statement of the number of revolutions per minute of the engines when the power is being measured, the revolutions per minute *at the time* the indicator is used. It is surprising how closely the curves of mean pressure referred to L.P. for a cargo-passenger steamer, a yacht, a tramp, a trawler, or a huge liner resemble those of "Argus" and "Edgewater" when plotted on percentage abscissæ of revolutions in this way. A curve from $(\text{revolutions})^3$ is a useful guide on such a diagram. Coal-consumption and steam-consumption results can be plotted and faired for ships just as well as results in land practice for consumption in lbs. per kilowatt hour, per B.H.P. hour, or per I.H.P. hour on abscissæ representing fractions of full load. The same method extended to propulsive efficiency and propeller efficiency is illustrated on Plate 37, where the curves marked C, E, G, H, K are taken from the interesting table of typical ships given in the excellent paper on "Geared Turbines for Ship Propulsion" to the Institution of Engineers and Shipbuilders in Scotland in 1914 by Messrs G. M. Welsh and W. D. M'Laren. Messrs M'Laren and Welsh gave the trial particulars. We have added the probable sea speeds.

C, Twin-screw Channel steamer with geared turbines. 324 × 40·5 × 12 ft. draught. Displacement = 2 380 tons. Block coefficient = ·53. Prismatic coefficient = ·57. Midship section coefficient = ·930. 22 knots on trial. 280 revolutions. 3 blades. Diameter = 9 ft. 6 in. Pitch = 10 ft. 0 in. 8 190 total S.H.P.

E, Single-screw cargo tramp, with either triple-expansion reciprocating steam engines or geared turbines. 400 × 53·4 × 25 ft. mean draught. Displacement = 11 900 tons. Block coefficient = ·78. Prismatic coefficient = ·81. Midship-area coefficient = ·963. 11 knots trial. 2 250 S.H.P. 70 revolutions. 4 blades. Diameter = 18 ft. 6 in. Pitch = 17 ft. 6 in. About 10·7 knots at sea with the same power (recip. 2 607 I.H.P.).

G, Twin-screw passenger and cargo vessel. Triple-expansion reciprocating steam engines. 484 × 60·5 × 19·4 ft. mean draught at trial. Displacement = 11 520 tons. Block coefficient = ·71. Prismatic coefficient = ·75. Midship-area coefficient = ·947. 16 knots on trial. 87 revolutions. 3 blades. Diameter = 17 ft. 3 in. Pitch = 21 ft. 6 in.

7 250 total I.H.P. About 15·6 knots at sea with the same power at the same draught, or about 15·15 knots at 24 ft. 6 in. draught at sea.

H, Twin-screw passenger liner, with quadruple-expansion reciprocating steam engines. $529 \times 62{\cdot}2 \times 21{\cdot}2$ ft. mean draught at trial. Displacement = 13 560 tons. Block coefficient = ·68. Prismatic coefficient = ·72. Midship-area coefficient = ·944. 18 knots on trial. 82 revolutions per minute. 4 blades. Diameter = 18 ft. 3 in. Pitch = 24 ft. 9 in. 10 930 total I.H.P. on trial. About 17·1 knots at sea with the same power at the same draught, or about 17 knots at sea on 25 ft. draught with the same power.

K, Quadruple-screw liner, with steam turbines direct coupled to propeller shafts. $729 \times 81 \times 29{\cdot}2$ ft. mean draught. Displacement = 29 550 tons. Block coefficient = ·60. Prismatic coefficient = ·64. Midship-section coefficient = ·938. 24 knots. 42 000 total S.H.P. 230 revolutions per minute. 4 blades. Diameter = 11 ft. 9 in. Pitch = 13 ft. 0 in.

The steam consumption in lbs. per H.P. hour are given on the same diagram as the propulsive coefficient and the propeller efficiency (Plate 37).

The loss of power by friction of the shafting and stern tube may be estimated at 4 per cent., and if the alignment is good this estimate is not far wrong. The efficiency of the gearing does not usually enter the power calculations, because it is understood that the rated S.H.P. is to be developed abaft the gear box, and steam consumption rates are always understood to be on this basis. The mechanical efficiency of a good double-reduction gear is about 95 per cent.

With regard to the propulsive coefficient $\left(\frac{\text{E.H.P.}}{\text{S.H.P.}}\right)$, when the S.H.P. is measured, as it usually is, aft of the thrust block, if we take Taylor's E.H.P.,

$$\text{S.H.P.} = \text{E.H.P.} \times \frac{1}{\text{Hull efficy.}} \times \frac{1}{\text{propeller efficy.}} \times \frac{1}{\text{transmission efficy.}}$$

$$= \text{E.H.P.} \times \frac{1}{\cdot 95} \times \frac{1}{\cdot 66} \times \frac{1}{\cdot 96}$$

$$= \frac{\text{E.H.P.}}{\cdot 60} \text{ for trial trip results,}$$

a higher propulsive coefficient than the average, but justified by results which have been analysed for passenger liners.

$\frac{\text{E.H.P. at } 14\frac{1}{2} \text{ knots (smooth water)}}{\text{I.H.P. at } 14\frac{1}{2} \text{ knots at sea}} = \cdot 45$ to $\cdot 50$ according to weather.

TABLE XLIV.—e_1, OR MECHANICAL EFFICIENCY OF MAIN ENGINES AT FULL POWER.

	Efficiency of gearing.	Thrust block.	Other losses or gains.	Ratio of power delivered to propeller to power at aft end of engine, or overall efficiency of the gearing at full power.
Geared turbines with mechanical gearing 20 : 1. Double reduction.	98 per cent. 95 per cent.	1 per cent. loss.	Windage, 2 or 3 per cent. loss due to astern turbine of 50 per cent. capacity. For higher astern power the windage loss is greater.	·94 or ·95 with astern turbine of half ahead power. ·92 or ·93 if astern turbine is for higher power.
Hydraulic transformer (Föttinger). Reduction ratio 4·5 : 1 to 10 : 1.	90 per cent.	Included in the foregoing.	2 per cent. gain due to making use of transformer waste heat in heating the feed water.	·92 large powers. ·88 small powers. ·90 possible astern.
Turbines with electrical transformers. Reduction ratio 18 : 1.	90 per cent. or less.	2 per cent.	Generators, motors, shafting, 10 per cent. loss or more.	·80 to ·88.
Direct turbines.	..	2 per cent.	1 per cent. loss in shafting.	·97, *i.e.* $\frac{\text{D.H.P.}}{\text{S.H.P.}} = \cdot 97$ = shaft transmission efficiency.
Reciprocating steam engines.	..	2 per cent.	Friction of engines and shafting 6 to 14 per cent. loss.	·83 to ·90, seldom over ·88.

There is a difference of about 13 or 14 per cent. in power for a difference of ½ a knot in speed between 14 and 15 knots, and

this is about the amount accounted for by ordinary moderately good weather and sea. Or we may take about 15 per cent. increase in S.H.P. at sea for corresponding speed on smooth water trial. Superintendent engineers should be able to get reliable figures for this, but power readings which appear consistent with the log book are difficult to find.

ELECTRIC TRANSMISSION.

From a diagram by Mr H. A. Mavor, printed for the discussion on Mr Bell's paper to the I.N.A. in 1908 on the trials of the "Lusitania," the following comparison was given, from particulars furnished by Messrs C. A. Parsons & Co. in 1907 :—

Shaft H.P.	10 000	20 000	30 000	40 000	50 000	60 000	70 000	75 000
"Lusitania," lbs. steam per S.H.P. hour.	24·8	19·7	15·5	14·3	13·5	13	12·6	12·4
"Carville," steam in lbs. per S.H.P. hour.	17	14	12·7	12·2	12	12	12·1	12·25

For direct comparison at relative fractions of full load, the Carville figures were adjusted by translating the kilowatts into shaft horse-power assumed to be delivered by a motor of 94 per cent. efficiency—*i.e.* 1 K.W. = 1·26 S.H.P.

The Carville trials were with superheated steam, and a correction of 1 per cent. for each 10° Fahr. of superheat was applied, so as to make the comparison as for saturated steam.

In the case of the Carville plant (a land installation), the efficiency was said to be maintained nearly constant up to double full load, the actual shaft H.P. being about one-fourteenth of that of "Lusitania."

Shaft Friction.—The following information has been taken from the *Transactions of the Institute of Marine Engineers*, December 1915 :—Tail shaft with brass liner, running in stern bush lined with lignum vitæ, and lubricated with sea water ; coefficient of friction = ·094. Steel shaft running in white metal, and lubricated with oil ; coefficient of friction = ·048.

Thrust-Block Friction.

Experiments have been made with marine engines to determine the amount of power lost by friction at the thrust block. In a paper on this subject read before the Institution of Naval Architects (see *Transactions*, 1899), Herr F. von Kodolitsch described how its amount had been electrically measured in the case of a triple-expansion marine engine of 600 I.H.P. The engine had cylinders $13\frac{7}{8}$ in. – $22\frac{1}{4}$ in. – and 36 in. dia. × 24 in. stroke. At 136 revolutions per minute, the speed of ship being 12 knots, I.H.P. = 600.

$$\frac{\text{Indicated amperes} \times \text{indicated volts}}{746} = \text{indicated electrical horse-power.}$$

With a thrust block of the ordinary type, 29·75 I.H.P. were lost; and with a thrust-block on the roller system, 2·4 horse-power were lost. Taking the first result as an average for ordinary marine engines, then we may say that $\frac{1}{20}$ or 5 per cent. of the indicated horse-power is lost in thrust-block friction. The Michell thrust block, with only one thrust shoe, used in most geared turbine steamers, minimises the friction loss.

Engineering, in an article dated 1st December 1916 on "The Willans Line for Steam Turbines," refers to the larger proportion of the wastes of energy, which are due mainly to windage, leakage losses, and fluid friction, as proportional to the load. "The resistances and losses which are independent of the load are merely those due to the bearings and thrust block, the oil pump and governor drive, and to the glands." Examples are given showing how the latter, "the constant losses, can be calculated with fair accuracy. The power absorbed in a turbine bearing in ordinary running conditions is, for example, given by the relation :—Power absorbed in bearing, d inches in diameter and l inches long,

$$= \frac{1}{3}\left(\frac{d}{10}\right)^2\left(\frac{l}{10}\right)\frac{\text{R.P.M.}}{100} \text{ in horse-power}$$

$$= \frac{1}{4}\left(\frac{d}{10}\right)^2\left(\frac{l}{10}\right)\frac{\text{R.P.M.}}{100} \text{ in kilowatts}$$

$$= 850\left(\frac{d}{10}\right)^2\frac{l}{10} \cdot \frac{\text{R.P.M.}}{100} \text{ in B.Th.U. per hour.}$$

The above coefficients, it will be seen, are rounded-off numbers, since 3 kw. is not exactly 4 h.p."

In a turbine having main bearings 12 in. in diameter by 38 in.

long, running at 750 r.p.m., the power absorbed by one bearing is 13·7 h.p. "The thrust bearing has 12 collars 1 in. deep and 10 in. in mean diameter. The resistance of such a thrust block is approximately ½ lb. for each square inch of oil under load, so that the power absorbed by the thrust block is in round figures given by the relation

$$\text{Power absorbed} = \frac{Nb}{8} \cdot \left(\frac{d}{10}\right)^2 \cdot \frac{\text{R.P.M.}}{100} \text{ horse-power,}$$

where N denotes the number of collars, b their breadth in inches, and d the mean diameter in inches of the collars. For the same turbine this formula gives 11·3 h.p. as the power absorbed. If we increase this to 16 h.p., the power absorbed by the oil pump and governor drive will be sufficiently allowed for."

The experiments of Gibson and Ryan on the friction of rotating discs (*Min. Proc. Inst. C.E.*, vol. clxxix) make possible a fair estimate of the power absorbed by the water glands. From these experiments we find that the power absorbed by a smooth thin disc of diameter d in., rotating in water, is given by the relation

$$\text{Friction H.P.} = \frac{1}{5\,000} \cdot \left(\frac{d}{10}\right)^2 \left[\frac{d}{10} \cdot \frac{\text{R.P.M.}}{100}\right]^{2\cdot 8}.$$

The addition of ribs to the disc increased this in a ratio of, say, 4·5 to 1.

From the above, and noting that a disc has two sides, we further deduce that the frictional resistance of a cylinder of diameter d and length l is expressed by

$$\text{Friction H.P.} = \frac{1}{2\,100} \cdot \frac{d}{10} \cdot \frac{l}{10}\left[\frac{d}{10} \cdot \frac{\text{R.P.M.}}{100}\right]^{2\cdot 8}.$$

In the turbine mentioned, 5 347 b.kw., the fixed resistances were estimated as 66 kw. "The indicated efficiency of a turbine varies with the ratio of expansion. In many cases, particularly with reaction turbines, which for commercial reasons have hitherto been run much below their most economical speed, the indicated efficiency at first increases as the load is reduced, afterwards diminishing again somewhat rapidly. It is, however, possible, to a fair degree of accuracy, to deduce from the actual Willans line the Willans line corresponding to a constant indicated efficiency by making use of the proposition, which is very approximately true, that when a turbine is throttle-governed the indicated efficiency depends solely on the ratio of initial to

final pressure." A diminution of vacuum from 28·51 to 27·23 in. would reduce the gross output from 5 413 to about 5 100 gross kw., and would increase the steam consumption per b.kw. hour by about 6 per cent.

Engine Efficiency.

(*a*) *Reciprocating Steam Engines.*—The indicated horse-power, as given by the indicator, exceeds the power delivered to the propeller by a considerable amount, on account of the friction of the moving parts, but by how much it is difficult to say definitely. The late Mr Blechynden's conclusions on the subject, published in the *Transactions of the North-East Coast Institution of Engineers and Shipbuilders*, 1891, are still of value, and are perhaps as sound as any that have since been promulgated.

$e = \frac{\text{S.H.P.}}{\text{I.H.P.}}$ = the mechanical efficiency of the engines, or "engine efficiency," the ratio of the work got out to the work put in.

Torsionmeters are rarely applied to reciprocating engines on account of the unevenness of the turning moment, the fluctuations in the readings being so great that it is seldom considered possible to obtain from them an accurate estimate of the shaft horse-power (S.H.P.). *The Shipbuilding and Shipping Record*, 16th July 1914, mentions, however, that Messrs Denny & Co. claim to have had fairly reliable readings with torsionmeters on reciprocating engines, and have arrived at the conclusion that it is not unusual to have engine efficiencies of 92 per cent. The North German Lloyd claimed 94 per cent. in one large steamer's engines. The friction of the engines and shafting consists of initial friction + load friction. In a progressive speed and power diagram, plotted upon speed in knots as abscissæ and horse-power as ordinates, the power expended in overcoming initial friction + load friction is represented by a slightly curved line, concave upwards (almost straight) below the I.H.P. curve, sloping gradually upwards from slow speeds to full speed. The power delivered to the propeller (*i.e.* as nearly as possible the brake horse-power at the propeller shaft) at any speed is the difference between the ordinate of this curve and that of the curve of I.H.P. A good example of curves of initial friction and load friction is to be found in Professor C. H. Peabody's paper to the American Society of Naval Architects and Marine Engineers, 1899, on the trials of U.S.S. "Manning," where the power expended on engine friction at full speed was 11·4 per cent. of the maximum I.H.P. The engine efficiency was therefore ·886. In the progressive trial, at

speeds varying from 5 knots to 16 knots, the engine efficiency varied from ·565 to ·886; in other words, the shaft horse-power varied from 56·5 per cent. to 88·6 per cent. of the indicated horse-power.

In a discussion at the Institution of Naval Architects in 1898, Sir Wm. H. White gave it as his opinion, resting on a large number of analyses, that, with a waste on the propeller of from 30 per cent. to 35 per cent., the dead load friction (or initial friction) might vary from 5 to 9 per cent., and the working load friction from 7 per cent. to 8 per cent. at full power, and that the delivery of power to the propellers at full power would therefore not be likely to exceed 80 per cent. to 85 per cent. of the I.H.P.

Mr D. W. Taylor gives initial friction about 3 to 9 per cent., depending upon the number of pumps worked off the main engine, and load friction about 7 per cent. of the rêmainder after deducting initial friction power from the original I.H.P. at full speed. By his focal-diagram method the initial friction has been very carefully computed for several vessels.

Our own opinion is that when only the air, feed, and bilge pumps are driven from the main engine levers, we may take the engine efficiency at about ·86 at sea for good engines, running at 600 to 700 feet per minute piston speed, and ·87 at maximum trial power. For engines driving reciprocating circulating pumps in addition to air, feed, and bilge pumps, the engine efficiency may be taken at ·84 at sea (*i.e.* at about ·9 of full power) and about ·85 to ·855 at maximum power. With all the pumps independent of the main engines, the mechanical efficiency may be ·87 on ordinary service at ·9 full power, and ·88 at maximum trial power; and with forced lubrication, as in some first-class cruisers completed in 1907, about ·89 to ·90. On the basis of trials of large vertical engines of the marine type driving electric generators, it is often assumed that the mechanical efficiency of the engine is ·86 to ·90 at full power. The Vulcan Company are said to have proved that the mechanical efficiency of the main engines of the "Kaiser Wilhelm II." was ·94 in ordinary service, but this is too high a figure to take as an average.

(*b*) *Mechanical Efficiency of Reciprocating Internal Combustion Engines.*—In most marine four-cycle motors driving an air compressor direct, and also with circulating water and lubricating pumps, the efficiency may be taken as from ·75 to ·80—·78 per cent. being perhaps a fair average. When the air compressor is not driven by the main engine, a higher efficiency may be obtained. In very exceptional cases it has reached ·85, but ·80 is a fairer figure to take as an average.

In two-cycle engines driving air compressors and one or two auxiliary pumps, the mechanical efficiency does not at present exceed about ·72. In determining the power for a motor-driven ship, 10 per cent. should be added if running in (tropical) waters over 80° F. (See p. 392.)

(*c*) *Geared Turbines.*—With good mechanical gearing the loss is very slight—perhaps 2 per cent., *i.e.* the mechanical efficiency is of the main engines, and gearing may be ·98, though it is often taken as ·95.

(*d*) *Direct Turbines.*—The S.H.P. by torsionmeter, the power delivered to the propeller, should be prized as an invaluable figure whenever it can be obtained. The torsionmeter can be used to determine the loss due to thrust-block friction.

In settling the horse-power required for a new ship, from model experiments, it is usual to take the E.H.P. obtained from a naked model, *i.e.* a model without appendages. The ratio of the E.H.P. from the naked model to the I.H.P. of the full-sized ship with appendages is, of course, a lower propulsive coefficient than the propulsive coefficient which would be obtained by using an E.H.P. obtained from a model with appendages; but this is largely due to the fact that the eddies for models with appendages differ from those of full-sized ships, and appendage resistance from models is apt to be exaggerated.

For ships driven by reciprocating steam engines $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ is frequently ·55, though ·50 is usually taken in design.

Corresponding to the figure given above, a lower figure, say ·44, should be taken when the propeller shafts are driven direct by turbines.

Analyses for a great many ships show a considerable variation in propulsive coefficients, but these are fairly consistent for types of ships, and all the small low-speed boats show low coefficients and the high-speed liners high coefficients. So far no conclusion has been arrived at as to why this should be so.

In the discussion on a paper by Mr T. G. Owens to the Inst. N.A. in 1914, the consistently high propulsive coefficients of vessels with triple screws as compared with quadruple screw ships was ascribed largely to the better utilisation of the wake. Signor Orlando remarked upon the inferior position, in that respect, of the wing propellers of the four-screw vessels, and the increase of resistance due to the appendages of the shafts.

Gunboat "Ceram." (*Trans. Inst. Naval Architects,* 1888.) Trials (July 26), 8·95 ft. mean draught. Copper sheathing. E.H.P. from model experiments. Actual ship :—152 × 25·6 × 8·95 ft. mean draught. Block coefficient = 0·513. Mid-area coefficient = 0·783. Mid area = 179 sq. ft. Prismatic coefficient = 0·654. Displacement = 510 tons. Wetted surface = 4 600. Cylinders $\frac{20\text{ in.} - 29\text{ in.} - 46\text{ in.}}{27\text{ in.}}$ × 120 lb. press. Propeller 4 blades. Diameter = 9 ft. Expanded surface = 30 sq. ft. Pitch = 13 ft.

Knots.	8·7	9·7	10·6	11·35	12
E.H.P.	118	169	230	298	372

E.H.P. calculated by author of paper.

Skin H.P.	61·2	83·7	107	130	151·6
Wave H.P.	56·8	85·3	123	168	220·4

Knots.	I.H.P.	Skin H.P.	$\frac{\text{E.H.P.}}{\text{I.H.P.}}$	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
6·25	61·2	24·1	...	254
8·5	197·4	57·5	·583	198
8·84	204·4	64	·59	215
8·95	219·5	66·2	·592	209
9·396	255·1	76	·60	206
12·124	607	156·3	·645	187
12·19	616·6	159	·645	186

I.H.P. varies as $(\text{speed})^4$ at 11·96 knots.

The coefficient of skin friction "f" is taken at 0·00953 for the full-sized ship.

Cruiser "Colorado." (*Proceedings American Society of Naval Architects and Marine Engineers*, 1904. Paper by Mr J. W. Powell.) Actual vessel:—502 × 69·5 × 23·92 ft. draught. Displacement = 13 670 tons. Block coefficient = 0·581. Wetted surface = 44 250 sq. ft. Midship area = 1 595·5 sq. ft. Mid-area coefficient = 0·972. Trials in 29 fathoms. Area of water line, 23 900 sq. ft. Coefficient of water plane = 0·688. Angle of W.L. entrance = 12°. Angle of run = 17·5°. Prismatic coefficient = 0·599.

Engines $\frac{38\frac{1}{2}\text{ in.} - 63\frac{1}{2}\text{ in.} - 74\text{ in.} - 74\text{ in.}}{48\text{ in.}} \times 265$ lbs. Heating surface = 68 537 sq. ft. Grate area = 1 632 sq. ft. Two propellers, three-bladed. Diameter = 18 ft. Pitch = 22 ft. 92 sq. ft. expanded surface each.

Knots.	Mean I.H.P.	E.H.P. from tank.	Revs.	Propeller effcy.	I.H.P. sq. ft. W.S.	App. slip per cent.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Skin H.P.	$\frac{\text{E.H.P.}}{\text{I.H.P.}}$
15·5	7 100	3 500	84	50	..	14·2	300	2 860	·494
17	8 800	4 700	91	54·5	..	14·3	320	3 710	·535
19	12 600	7 000	103	56	..	14·8	312	5 110	·555
20	16 000	8 600	109	54	..	15·4	286	5 910	·587
21	20 300	10 900	115	54	..	16·3	261	6 770	·537
22	24 100	13 800	122·3	57·5	·545	17·5	253	7 750	·573
22·24	25 000	14 500	124	58	..	17·8	252	7 980	·58

100-ft. model of "Colorado":—100 × 13·85 × 4·77 ft. draught. Displacement = 108 tons. Block coefficient = 0·581. W.S. = 1 757. Mid-area coefficient = 0·972. Prismatic coefficient = 0·599.

Humps and hollows clearly marked.

U.S.S. "Manning." Single-screw. (Described by Professor Cecil H. Peabody in *Proceedings of the American Society of Naval Architects and Marine Engineers.*) Actual ship:—188 × 32·81 × 12·33 ft. mean draught. Displacement = 1 000·7 tons. Block coefficient = 0·48. Wetted surface = 7 273 sq. ft.

Engines, $\frac{25 \text{ in.} - 37\frac{1}{2} \text{ in.} - 56\frac{1}{4} \text{ in.}}{30 \text{ in.}}$. Propeller diameter = 11 ft.

Pitch = 12·33 ft. $\frac{\text{Pitch}}{\text{Diameter}} = 1{\cdot}121$. Area ratio = 0·421. Hub. = 1·875 ft. diameter.

Knots.	I.H.P.	Revs.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	T.H.P.	Initial friction power.	Load friction power.	Skin resistance power.	Wave resistance power.	Wake gain and thrust deduction.	Engine efficiency.
5	69	42·8	180	30	27	3	20	5	5	·565
6	100	51·5	215	48	33	5	34	7	7	
7	141	60·1	243	74	38	7	52	11	11	·68
8	194	68·8	264	108	44	10	76	16	16	·744
9	263	77·4	276	153	49	15	106	24	23	·757
10	354	86·3	283	214	55	21	142	40	82	·794
11	486	95·8	274	304	61	30	187	71	46	·812
12	671	106·2	257	431	68	42	239	127	65	·836
13	920	116·7	238	600	74	59	299	211	90	855
14	1 245	127·7	220	820	81	81	369	328	127	·87
15	1 661	139·5	203	930	89	110	449	481	160	·88
16	2 181	152	188	1 221	97	146	539	682	214	·886

The I.H.P. varies as the fourth power of the speed at about 15·1 knots.

Notice that T.H.P. (thrust horse-power) = Skin horse-power + Wave-making horse-power + wake gain and thrust deduction power.

And E.H.P. (effective horse-power) = Skin friction horse-power + Wave-making horse-power.

Torpedo-boat "Biddle." Twin-screw. (From the *Proceedings of the American Society of Naval Architects and Marine Engineers.* Paper by Mr. Chas. P. Wetherbee.) Actual vessel :—157 × 16·25 × 4·81 ft. mean draught. 4·4 tons per in. immersion. Wetted surface = 2 540 sq. ft. 168 tons displacement. Block coefficient = 0·478. Mid-area coefficient = 0·724. Coefficient water plane (on trial) = 0·743. Prismatic coefficient = 0·663. Propellers, diameter = 6·68 ft. Pitch = 10·88 ft. Projected surface each = 1 440 sq. in.

Progressive trial of T.B. "Biddle," clean bottom, 35 days afloat.									Trial of "Barney," sister ship (two boats identical), dirty bottom, 126 days out.	
Knots.	Revs.	App. slip per cent.	I.H.P.	Wave H.P.	Skin H.P.	E.H.P.	Propulsive coefficient.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Revs.	I.H.P.
11	117	12·61	220	30	65	95	·432	183·4	...	...
13	137	11·79	355	75	105	180	·507	187·6	137·4	396
15	158	11·75	522	130	160	290	·556	196	160·5	602
17	181·5	12·93	760	245	225	470	·618	196	185·2	927
18	194·5	13·97	928	325	265	590	·635	190·5	198	1 150
19	207·7	14·97	1 138	420	305	725	·637	182·7	210·9	1 410
20	220	15·49	1 370	500	355	855	·624	177	223·4	1 705
21	231·4	15·64	1 600	585	405	990	·619	175	235·2	2 002
22	242	15·49	1 835	665	465	1 130	·616	175·9	246	2 290
23	252·4	15·29	2 080	750	530	1 280	·615	177	256·6	2 585
24	262·6	15·04	2 346	840	590	1 430	·610	178	266·7	2 892
25	273	14·87	2 636	915	670	1 585	·601	179	277	3 230
26	283·3	14·68	2 932	1 005	740	1 745	.595	182	287·5	3 590
27	294	14·63	3 257	1 080	830	1 910	·586	183	298	3 960
28	304·2	14·44	3 572	1 165	920	2 085	·583	186	308·4	4 340
29	314·8	14·37	3 910	1 255	1 015	2 270	·581	189	318·2	4 730
30	325·2	14·24	4 225	1 340	1 120	2 460	·582	193	...	...

The I.H.P. is varying as the 3·2 power of the speed at about 28·8 knots.

TABLE XLV.—TWO-THIRDS POWERS OF NUMBERS.

Number.	$\frac{2}{3}$rd power.	Number.	$\frac{2}{3}$rd power.	Number.	$\frac{2}{3}$rd power.	Number.	$\frac{2}{3}$rd power.
		41	11·9	81	18·72	310	45·80
2	1·58	42	12·1	82	18·87	320	46·78
3	2·08	43	12·27	83	19·05	330	47·75
4	2·519	44	12·48	84	19·2	340	48·71
5	2·924	45	12·65	85	19·31	350	49·66
6	3·302	46	12·85	86	19·45	360	50·61
7	3·659	47	13·03	87	19·65	370	51·54
8	4·00	48	13·2	88	19·8	380	52·46
9	4·326	49	13·4	89	19·95	390	53·38
10	4·641	50	13·58	90	20·1	400	54·29
11	4·946	51	13·75	91	20·25	410	55·19
12	5·241	52	13·93	92	20·4	420	56·08
13	5·528	53	14·11	93	20·52	430	56·97
14	5·808	54	14·3	94	20·66	440	57·85
15	6·082	55	14·46	95	20·81	450	58·72
16	6·349	56	14·65	96	20·95	460	59·59
17	6·611	57	14·8	97	21·1	470	60·45
18	6·868	58	14·98	98	21·25	480	61·30
19	7·12	59	15·15	99	21·4	490	62·15
20	7·368	60	15·33	100	21·54	500	62·99
21	7·611	61	15·5	110	22·96	510	63·83
22	7·851	62	15·68	120	24·33	520	64·66
23	8·087	63	15·83	130	25·66	530	65·49
24	8·320	64	16·0	140	26·96	540	66·31
25	8·549	65	16·17	150	28·23	550	67·13
26	8·776	66	16·35	160	29·47	560	67·94
27	9·00	67	16·5	170	30·69	570	68·74
28	9·22	68	16·67	180	31·88	580	69·54
29	9·439	69	16·83	190	33·05	590	70·34
30	9·654	70	16·98	200	34·21	600	71·13
31	9·868	71	17·15	210	35·33	610	71·92
32	10·08	72	17·3	220	36·44	620	72·71
33	10·28	73	17·46	230	37·54	630	73·49
34	10·49	74	17·67	240	38·62	640	74·26
35	10·70	75	17·8	250	39·68	650	75·03
36	10·90	76	17·93	260	40·74	660	75·80
37	11·10	77	18·1	270	41·78	670	76·57
38	11·30	78	18·25	280	42·80	680	77·33
39	11·5	79	18·41	290	43·81	690	78·08
40	11·7	80	18·55	300	44·81	700	78·84

Table XLV.—Two-Thirds Powers of Numbers—*continued.*

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
710	79·59	1 110	107·20	1 510	131·61	1 910	153·94
720	80·33	1 120	107·85	1 520	132·19	1 920	154·47
730	81·07	1 130	108·49	1 530	132·77	1 930	155·01
740	81·81	1 140	109·13	1 540	133·35	1 940	155·54
750	82·55	1 150	109·76	1 550	133·93	1 950	156·08
760	83·28	1 160	110·40	1 560	134·50	1 960	156·61
770	84·01	1 170	111·03	1 570	135·08	1 970	157·14
780	84·73	1 180	111·67	1 580	135·65	1 980	157·68
790	85·4	1 190	112·30	1 590	136·23	1 990	158·21
800	86·18	1 200	112·92	1 600	136·80	2 000	158·74
810	86·89	1 210	113·55	1 610	137·37	2 020	159·79
820	87·61	1 220	114·17	1 620	137·93	2 040	160·84
830	88·32	1 230	114·80	1 630	138·50	2 060	161·89
840	89·03	1 240	115·42	1 640	139·06	2 080	162·94
850	89·73	1 250	116·04	1 650	139·63	2 100	163·99
860	90·43	1 260	116·66	1 660	140·19	2 120	165·02
870	91·13	1 270	117·27	1 670	140·75	2 140	166·05
880	91·83	1 280	117·89	1 680	141·32	2 160	167·09
890	92·52	1 290	118·50	1 690	141·88	2 180	168·12
900	93·22	1 300	119·11	1 700	142·44	2 200	169·15
910	93·91	1 310	119·72	1 710	143·00	2 220	170·17
920	94·59	1 320	120·33	1 720	143·55	2 240	171·19
930	95·28	1 330	120·94	1 730	144·11	2 260	172·20
940	95·96	1 340	121·55	1 740	144·66	2 280	173·22
950	96·64	1 350	122·15	1 750	145·22	2 300	174·24
960	97·32	1 360	122·75	1 760	145·77	2 320	175·24
970	97·99	1 370	123·35	1 770	146·32	2 340	176·25
980	98·66	1 380	123·95	1 780	146·87	2 360	177·25
990	99·33	1 390	124·55	1 790	147·42	2 380	178·26
1 000	100·00	1 400	125·14	1 800	147·97	2 400	179·26
1 010	100·66	1 410	125·74	1 810	148·52	2 420	180·25
1 020	101·33	1 420	126·33	1 820	149·06	2 440	181·24
1 030	101·99	1 430	126·92	1 830	149·61	2 460	182·23
1 040	102·65	1 440	127·51	1 840	150·15	2 480	183·22
1 050	103·30	1 450	128·10	1 850	150·70	2 500	184·20
1 060	103·96	1 460	128·69	1 860	151·24	2 520	185·18
1 070	104·61	1 470	129·28	1 870	151·78	2 540	186·16
1 080	105·26	1 480	129·87	1 880	152·32	2 560	187·14
1 090	105·91	1 490	130·45	1 890	152·86	2 580	188·11
1 100	106·56	1 500	131·03	1 900	153·40	2 600	189·08

TABLE XLV.—TWO-THIRDS POWERS OF NUMBERS—*continued.*

Number.	$\frac{2}{3}$rd power.	Number.	$\frac{2}{3}$rd power.	Number.	$\frac{2}{3}$rd power.	Number.	$\frac{2}{3}$rd power.
2 620	190·05	3 420	226·99	4 220	261·14	5 050	294·34
2 640	191·02	3 440	227·88	4 240	261·96	5 100	296·27
2 660	191·98	3 460	228·76	4 260	262·78	5 150	298·21
2 680	192·93	3 480	229·64	4 280	263·60	5 200	300·15
2 700	193·89	3 500	230·52	4 300	264·42	5 250	302·06
2 720	194·85	3 520	231·40	4 320	265·24	5 300	303·98
2 740	195·80	3 540	232·27	4 340	266·06	5 350	305·89
2 760	196·75	3 560	233·14	4 360	266·87	5 400	307·80
2 780	197·71	3 580	234·02	4 380	267·69	5 450	309·68
2 800	198·66	3 600	234·89	4 400	268·51	5 500	311·58
2 820	199·60	3 620	235·76	4 420	269·32	5 550	313·46
2 840	200·54	3 640	236·62	4 440	270·13	5 600	315·34
2 860	201·48	3 660	237·49	4 460	270·95	5 650	317·21
2 880	202·42	3 680	238·36	4 480	271·76	5 700	319·09
2 900	203·35	3 700	239·22	4 500	272·56	5 750	320·95
2 920	204·28	3 720	240·08	4 520	273·37	5 800	322·81
2 940	205·22	3 740	240·98	4 540	274·17	5 850	324·66
2 960	206·15	3 760	241·80	4 560	274·98	5 900	326·51
2 980	207·08	3 780	242·65	4 580	275·78	5 950	328·35
3 000	208·01	3 800	243·51	4 600	276·58	6 000	330·19
3 020	208·93	3 820	244·36	4 620	277·39	6 050	332·02
3 040	209·85	3 840	245·22	4 640	278·19	6 100	333·85
3 060	210·76	3 860	246·07	4 660	278·99	6 150	335·67
3 080	211·68	3 880	246·97	4 680	279·78	6 200	337·49
3 100	212·59	3 900	247·76	4 700	280·58	6 250	339·30
3 120	213·51	3 920	248·61	4 720	281·38	6 300	341·11
3 140	214·42	3 940	249·45	4 740	282·17	6 350	342·91
3 160	215·33	3 960	250·29	4 760	282·96	6 400	344·71
3 180	216·24	3 980	251·41	4 780	283·76	6 450	346·50
3 200	217·15	4 000	251·98	4 800	284·55	6 500	348·29
3 220	218·05	4 020	252·82	4 820	285·33	6 550	350·07
3 240	218·95	4 040	253·65	4 840	286·11	6 600	351·85
3 260	219·85	4 060	254·49	4 860	286·90	6 650	353·62
3 280	220·75	4 080	255·33	4 880	287·68	6 700	355·39
3 300	221·65	4 100	256·16	4 900	288·47	6 750	357·16
3 320	222·54	4 120	257·00	4 920	289·26	6 800	358·93
3 340	223·44	4 140	257·83	4 940	290·05	6 850	360·68
3 360	224·34	4 160	258·67	4 960	290·84	6 900	362·43
3 380	225·22	4 180	259·49	4 980	291·62	6 950	364·18
3 400	226·11	4 200	260·31	5 000	292·40	7 000	365·93

TABLE XLV.—TWO-THIRDS POWERS OF NUMBERS—*continued.*

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
7 050	367·67	9 050	434·27	12 100	527·05	16 100	637·6
7 100	369·41	9 100	435·86	12 200	529·95	16 200	640·1
7 150	371·13	9 150	437·45	12 300	532·83	16 300	642·9
7 200	372·86	9 200	439·04	12 400	535·72	16 400	645·4
7 250	374·58	9 250	440·64	12 500	538·60	16 500	648·1
7 300	376·31	9 300	442·23	12 600	541·48	16 600	650·6
7 350	378·02	9 350	443·82	12 700	544·34	16 700	653·2
7 400	379·74	9 400	445·40	12 800	547·20	16 800	655·9
7 450	381·44	9 450	446·97	12 900	550·04	16 900	658·5
7 500	383·15	9 500	448·54	13 000	552·88	17 000	661·1
7 550	384·85	9 550	450·11	13 100	555·70	17 100	663·7
7 600	386·55	9 600	451·68	13 200	558·53	17 200	666·2
7 650	388·24	9 650	453·25	13 300	561·35	17 300	668·9
7 700	389·93	9 700	454·82	13 400	564·16	17 400	671·4
7 750	391·62	9 750	456·39	13 500	566·96	17 500	674·0
7 800	393·30	9 800	457·95	13 600	569·76	17 600	676·5
7 850	394·98	9 850	459·50	13 700	572·54	17 700	679·1
7 900	396·66	9 900	461·06	13 800	575·33	17 800	681·6
7 950	398·33	9 950	462·61	13 900	578·10	17 900	684·2
8 000	400·00	10 000	464·16	14 000	580·88	18 000	686·8
8 050	401·66	10 100	467·25	14 100	583·63	18 100	689·3
8 100	403·32	10 200	470·33	14 200	586·38	18 200	691·9
8 150	404·97	10 300	473·39	14 300	589·13	18 300	694·4
8 200	406·63	10 400	476·44	14 400	591·88	18 400	696·9
8 250	408·28	10 500	479·49	14 500	594·61	18 500	699·5
8 300	409·93	10 600	482·54	14 600	597·34	18 600	702·0
8 350	411·57	10 700	485·57	14 700	600·07	18 700	704·5
8 400	413·22	10 800	488·60	14 800	602·80	18 000	707·0
8 450	414·85	10 900	491·61	14 900	605·51	18 900	709·5
8 500	416·49	11 000	494·61	15 000	608·22	19 000	712·1
8 550	418·12	11 100	497·60	15 100	610·91	19 100	714·6
8 600	419·75	11 200	500·58	15 200	613·57	19 200	717·0
8 650	421·37	11 300	503·56	15 300	616·22	19 300	719·5
8 700	423·00	11 400	506·53	15 400	619·00	19 400	722·0
8 750	424·62	11 500	509·48	15 500	621·6	19 500	724·5
8 800	426·24	11 600	512·43	15 600	624·3	19 600	727·0
8 850	427·85	11 700	515·38	15 700	626·9	19 700	729·4
8 900	429·46	11 800	518·31	15 800	629·6	19 800	731·9
8 950	431·06	11 900	521·23	15 900	632·2	19 900	734·4
9 000	432·67	12 000	524·15	16 000	634·9	20 000	736·8

Table XLV.—Two-Thirds Powers of Numbers—*continued.*

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
20 100	739 3	24 100	834·3	28 100	924·4	32 100	1 010·0
20 200	741·9	24 200	836·6	28 200	926·5	32 200	1 012·0
20 300	744·2	24 300	838·9	28 300	928·6	32 300	1 014·2
20 400	746·6	24 400	841·2	28 400	930·9	32 400	1 016·3
20 500	749·1	24 500	843·6	28 500	933·1	32 500	1 018·4
20 600	751·5	24 600	845·9	28 600	935·1	32 600	1 020
20 700	753·9	24 700	848·1	28 700	937·4	32 700	1 022
20 800	756·4	24 800	850·5	28 800	939·6	32 800	1 024
20 900	758·7	24 900	852·9	28 900	941·9	32 900	1 026
21 000	761·1	25 000	855·0	29 000	944·0	33 000	1 028
21 100	763·9	25 100	857·3	29 100	946·1	33 100	1 030
21 200	766·0	25 200	859·6	29 200	948·3	33 200	1 033
21 300	768·4	25 300	861·9	29 300	950·4	33 300	1 035
21 400	770·7	25 400	864·1	29 400	952·6	33 400	1 037
21 500	773·4	25 500	866·3	29 500	954·9	33 500	1 039
21 600	775·6	25 600	868·6	29 600	956·9	33 600	1 041
21 700	778·0	25 700	870·9	29 700	959·0	33 700	1 043
21 800	780·3	25 800	873·1	29 800	961·3	33 800	1 045
21 900	782·8	25 900	875·4	29 900	963·3	33 900	1 047
22 000	785·2	26 000	877·7	30 000	965·4	34 000	1 049
22 100	787·5	26 100	880·0	30 100	967·6	34 100	1 051
22 200	789·9	26 200	882·1	30 200	969·7	34 200	1 053
22 300	792·2	26 300	884·4	30 300	971·9	34 300	1 055
22 400	794·6	26 400	886·6	30 400	974·0	34 400	1 057
22 500	797·0	26 500	888·9	30 500	976·2	34 500	1 059
22 600	799·4	26 600	891·0	30 600	978·3	34 600	1 061
22 700	801·9	26 700	893·4	30 700	980·4	34 700	1 063
22 800	804·0	26 800	895·5	30 800	982·5	34 800	1 065
22 900	806·4	26 900	897·8	30 900	984·6	34 900	1 068
23 000	808·8	27 000	900·0	31 000	986·8	35 000	1 070
23 100	811·1	27 100	902·2	31 100	988·9	35 100	1 072
23 200	813·4	27 200	904·4	31 200	991·1	35 200	1 074
23 300	815·8	27 300	906·6	31 300	993·1	35 300	1 076
23 400	818·1	27 400	908·9	31 400	995·2	35 400	1 078
23 500	820·4	27 500	911·1	31 500	997·4	35 500	1 080
23 600	822·8	27 600	913·3	31 600	999·5	35 600	1 082
23 700	825·1	27 700	915·5	31 700	1 001·6	35 700	1 084
23 800	827·4	27 800	917·5	31 800	1 003·7	35 800	1 086
23 900	829·7	27 900	919·9	31 900	1 005·8	35 900	1 088
24 000	832·0	28 000	922·1	32 000	1 007·9	36 000	1 090

TABLE XLV.—TWO-THIRDS POWERS OF NUMBERS—*continued.*

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
36 100	1 092	40 100	1 171	44 100	1 248	48 100	1 323
36 200	1 094	40 200	1 173	44 200	1 250	48 200	1 324
36 300	1 096	40 300	1 175	44 300	1 252	48 300	1 326
36 400	1 098	40 400	1 177	44 400	1 254	48 400	1 328
36 500	1 100	40 500	1 179	44 500	1 255	48 500	1 330
36 600	1 102	40 600	1 181	44 600	1 257	48 600	1 332
36 700	1 104	40 700	1 183	44 700	1 259	48 700	1 334
36 800	1 106	40 800	1 185	44 800	1 261	48 800	1 335
36 900	1 108	40 900	1 187	44 900	1 263	48 900	1 337
37 000	1 110	41 000	1 189	45 000	1 265	49 000	1 339
37 100	1 112	41 100	1 190	45 100	1 267	49 100	1 341
37 200	1 114	41 200	1 192	45 200	1 269	49 200	1 343
37 300	1 116	41 300	1 194	45 300	1 271	49 300	1 344
37 400	1 118	41 400	1 196	45 400	1 273	49 400	1 346
37 500	1 120	41 500	1 198	45 500	1 275	49 500	1 348
37 600	1 122	41 600	1 200	45 600	1 276	49 600	1 350
37 700	1 124	41 700	1 202	45 700	1 278	49 700	1 352
37 800	1 126	41 800	1 204	45 800	1 280	49 800	1 354
37 900	1 128	41 900	1 206	45 900	1 282	49 900	1 355
38 000	1 130	42 000	1 208	46 000	1 284	50 000	1 357
38 100	1 132	42 100	1 210	46 100	1 286	50 100	1 359
38 200	1 134	42 200	1 211	46 200	1 287	50 200	1 361
38 300	1 136	42 300	1 213	46 300	1 289	50 300	1 363
38 400	1 138	42 400	1 215	46 400	1 291	50 400	1 364
38 500	1 140	42 500	1 217	46 500	1 293	50 500	1 366
38 600	1 142	42 600	1 219	46 600	1 295	50 600	1 368
38 700	1 144	42 700	1 221	46 700	1 296	50 700	1 370
38 800	1 146	42 800	1 223	46 800	1 298	50 800	1 372
38 900	1 148	42 900	1 225	46 900	1 300	50 900	1 374
39 000	1 150	43 000	1 227	47 000	1 302	51 000	1 375
39 100	1 152	43 100	1 229	47 100	1 304	51 100	1 377
39 200	1 154	43 200	1 231	47 200	1 306	51 200	1 379
39 300	1 155	43 300	1 232	47 300	1 308	51 300	1 381
39 400	1 157	43 400	1 234	47 400	1 310	51 400	1 383
39 500	1 159	43 500	1 237	47 500	1 312	51 500	1 384
39 600	1 161	43 600	1 239	47 600	1 313	51 600	1 386
39 700	1 163	43 700	1 241	47 700	1 315	51 700	1 388
39 800	1 165	43 800	1 242	47 800	1 317	51 800	1 390
39 900	1 167	43 900	1 244	47 900	1 319	51 900	1 391
40 000	1 169	44 000	1 246	48 000	1 321	52 000	1 393

TABLE XLV.—TWO-THIRDS POWERS OF NUMBERS—*continued.*

Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.	Number.	⅔rd power.
52 100	1 395	54 600	1 439	57 100	1 483	59 600	1 526
52 200	1 397	54 700	1 441	57 200	1 485	59 700	1 528
52 300	1 398	54 800	1 443	57 300	1 486	59 800	1 530
52 400	1 400	54 900	1 445	57 400	1 488	59 900	1 531
52 500	1 402	55 000	1 446	57 500	1 490	60 000	1 533
52 600	1 404	55 100	1 448	57 600	1 492	60 100	1 534
52 700	1 406	55 200	1 450	57 700	1 493	60 200	1 536
52 800	1 407	55 300	1 452	57 800	1 495	60 300	1 538
52 900	1 409	55 400	1 453	57 900	1 497	60 400	1 539
53 000	1 411	55 500	1 455	58 000	1 499	60 500	1 541
53 100	1 413	55 600	1 457	58 100	1 500	60 600	1 543
53 200	1 414	55 700	1 458	58 200	1 502	60 700	1 545
53 300	1 416	55 800	1 460	58 300	1 504	60 800	1 546
53 400	1 418	55 900	1 462	58 400	1 506	60 900	1 548
53 500	1 420	56 000	1 464	58 500	1 508	61 000	1 550
53 600	1 422	56 100	1 466	58 600	1 509		
53 700	1 423	56 200	1 467	58 700	1 510		
53 800	1 425	56 300	1 469	58 800	1 512		
53 900	1 427	56 400	1 471	58 900	1 514		
54 000	1 429	56 500	1 473	59 000	1 516		
54 100	1 430	56 600	1 475	59 100	1 517		
54 200	1 432	56 700	1 476	59 200	1 519		
54 300	1 434	56 800	1 478	59 300	1 521		
54 400	1 436	56 900	1 480	59 400	1 523		
54 500	1 438	57 000	1 482	59 500	1 524		

I.

Simplified Ship Forms.—"Comparative Resistance of 'Ordinary Ship-shape' and 'Straight-Frame' Models." A paper by Professor H. C. Sadler and Mr T. Yamamoto, read at the Society of Naval Architects and Marine Engineers, Philadelphia, reprinted in *International Marine Engineering*, March 1919, gives an account of some experiments upon "straight-frame" forms conducted in the tank at the University of Michigan. Plans show the "straight-frame" form referred to. The models were 10 ft. long × 16 in. beam. The resistances were measured at three different draughts, 7 in., 6 in., and 5 in. For each type the following characteristics were kept constant, viz. length, breadth, draught, displacement (at load-draught with the corner cut off), the curve of sectional areas (and hence prismatic coefficient), and the shape of the water-line. Of the numerous forms tried, we select the two named Y. 1 C. and Y. 3 C. The differences in results are slight.

Draught.	B/*d*.	Coefficients.	Y. 1 C., corner off.	Y. 3 C.
in.		Longitudinal	·801	·798
7	2·285	Block	·779	·779
		Midship	·973	·976
		Longitudinal	·791	·788
6	2·66	Block	·766	·766
		Midship	·968	·972
		Longitudinal	·780	·778
5	3·2	Block	·749	·750
		Midship	·961	·965

II.

The effect of retaining the corner volume at the bilge was to increase the resistance about 3 to 4 per cent., or approximately the same as that due to the added surface. Compared with the ship-shape form, there was practically no difference in resistance between this and the simplified form with the corner cut off. Other varieties showed, at the lower speed-length ratios, little if any differences, and, such as there were, of the order of 1 to 2 per cent., while the effect of retaining the sharp corner appeared to increase the resistance, *i.e.* the resistance increased at a somewhat more rapid ratio than the added wetted surface.

The effect of the sharp corner upon the reduction of rolls was most marked, and even with the corner removed these models came to rest quicker than the ship-shape form.

The conclusions were :—

(1) Vessels of the straight-frame type can be designed which will have about the same resistance as a ship-shape form.

(2) If the diagonal line of the corner be given the wrong slope, this will increase the resistance due to the lack of conformity with the proper stream-line flow.

(3) The effect of maintaining the square corner is to increase the bare hull resistance, but as vessels of this form would not need bilge keels, the net result from a horse-power standpoint would be about the same as for a ship-shape form.

(4) Probably the best results from a resistance standpoint would be obtained by using diagonal line which is of a curved form in the body plan.

Straight-frame forms were discussed at the spring meeting of the Institution of Naval Architects, 1919, and it was pointed out that there was little to be gained from the point of view of simplicity in construction over the usual rounded bilge form, which was more adaptable and easily maintained.

Name.	Tons displacement.	Length B.P.	Beam.	Mean draught.	Block coef.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Sea speed in knots.	I.H.P.	Date.	Tons per inch.	Propellers. D.	P.	F.S.	No. of blades.	Revs. per min.
Francis	9 120	355	49·25	23·6	..	255	10·7	2 100	..	34·55	17	17	91	4	73
Cuthbert	8 975	355·1	49·3	23·6	..	253	10·7	2 100	..	34·45	17	16	86	4	66
Justin	8 930	355	48·7	23·6	..	252	10·7	2 100	..	34·35	16·75	17·5	84	4	72
Amazonense	6 035	312	40·9	22·9	..	243	9·3	1 100	..	25·75	15·25	17·75	70	4	57
Javary	2 750	235·5	34·3	15·11	..	220	9·0	650	..	16·75	12·8	15	55	4	69
Ucayali	2 262	230	32·2	15	..	212	9·5	700	..	15	13·5	16·5	54	4	65
Napo	2 445	224·4	33·4	15·4½	..	220	9·0	650	..	15·8	12·25	15·8	50	4	66
Dunstan	6 740	322	42·3	22·4	·778	..	10·5	1 350	..	27·5	15·5	16·5	72·3	4	70
Basil	7 495	338	43·7	23·6	..	278	10·5	1 600	..	29·2	16·5	18	74·8	4	65
Benedict	8 180	345	43·5	22·9½	..	266	9·5	1 450	..	29	16·75	17	75	4	66
Gregory	4 622	265	40	20·3	..	232	9·5	1 000	..	21·5	14·75	15·75	65	4	66
Augustine	6 558	359·6	43·8	23·5	..	255	12·5	2 700	..	28·7	18·5	25·5	87·5	4	57
Atahualpa	3 374	261·4	36·2	18	..	202	10·25	1 200	1894	18·3	14·2	15·5	63	4	70
Clement	7 027	345·7	44·1	23·1½	..	288	12·0	2 100	1896	30·2	17	23	80	4	62
Ambrose	7 654	375·2	47·8	23·5	..	303	14·5	3 900	1903	31·8	19	19	100	4	78
Anselm	9 170	400·4	50·1	23·5	..	300	14·0	4 000	1905	38·0	19	20·5	100	4	74
T.S.S. Antony *	9 460	418·5	52·3	23·5	..	270	14·0	4 500	1907	39·4	16·5	21·5	76	3	73
Manco	5 008	300·3	45·2	18	..	285	11·5	1 900	1908	26·0	15·5	14·3	72	4	88
T.S.S. Hilary *	9 300	418·5	52·2	23·5	..	270	14·0	4 500	1908	40·32	16·75	21·5	76	3	74
T.S.S. Hildebrand *	10 195	440·3	54·1	23·5	..	266	14·6	5 500	1911	43·15	16·75	18·7	72	3	87
Aidan	9 980	375·9	50·3	23·5½	·788	285	10·9	2 100	1911	38·0	17·5	15·5	95	4	72
Denis	9 982	376·4	50·3	23·6	..	285	10·9	2 100	1910	38	17	16	86	4	72
Christopher	8 978	360	50·1	23·1¾	..	273	10·8	2 000	1910	34	17·25	18·5	90	4	66
Alban	11 060	375	51·7	26·2	·764	275	11·0	2 400	1914	38·2	17·5	16·0	95	4	75

* All single screw except three marked T.S.S.

ACTUAL TWIN-SCREW VESSELS 200 TO 300 FEET IN LENGTH.

Name.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	Type of engines.	
U.S. coast-defence ship Monterey	4 084	256	59	14·75	5 072	14·4	150	1898	Recip. steam	3 blades. Dia. = 10·16′. Pitch = 11·66′. Pitch ratio = 1·15. Area = 38·9 sq. ft. Area ratio = ·480. Revs. = 162. App. slip per cent. = 27·1. (Durand.)
Ozark . . .	3 275	252	50·0	12·583	2 190	12·71	206·7	..	..	3 blades. D. = 9·0′. P. = 7·25′. Proj. area ratio = ·355. App. slip per cent. = 9·94. Revs. = 214·5. E.H.P. ÷ I.H.P. = 54·3 per cent. bare hull, and 63·37 with appendages. (Dyson.)
U.S. gunboat Nashville	1 364	221	38·0	10·95	2 489	16·3	214	..	Recip. steam	3 blades. D. = 6·67′. P. = 7·0′. Pitch ratio = 1·05. Area ratio = ·373. Revs. = 308·4. Area = 13·0. App. slip per cent. = 23·5. (Durand.)
U.S. gunboat Wilmington	1 342	250	39·6	8·65	1 858	15·08	223	1895	..	3 blades. D. = 7·0′. P. = 7·13′. Pitch ratio = 1·02. Area = 18·7. Area ratio = ·486. Revs. = 273. App. slip per cent. = 21·5. (Durand.)
Katahdin . .	2 139	250·2	41·67	14·90	4 904	16·07	141	..	..	3 blades. D. = 12·0′. P. = 14·0′. Pitch ratio = 1·17. Area = 36·0. Area ratio = ·318.

I.H.P.	Revs.	App. slip per cent.	Knots.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
4 904	139·5	16·55	16·07	141
4 618	137·7	16·7	15·84	143
3 040	126·25	14·75	14·87	180

TWIN-SCREW VESSELS 200 TO 300 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Block coefficient.	Midship-area coefficient.	Prismatic coefficient.
U.S. coast-defence ship Monterey	·642	·906	·709
Ozark	·719	·955	
U.S. gunboat Nashville	·518	·889	·583
,, ,, Wilmington	·549	·957	·574
Katahdin	·482	·746	·646

ACTUAL TWIN-SCREW VESSELS 200 TO 300 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam.	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{power}}$	Date	Type of engines.	
U.S. gunboat Yorktown .	1 680	228	36·0	13·84	3 579	16·65	182	..	Recip. steam	3 blades. Dia. = 10·50′. Pitch = 12·50′. Pitch ratio = 1·19. Area = 25·4. Area ratio = ·294. Revs. = 160·75. App. slip per cent. = 16·05. (Durand.)
„ „ Helena .	1 340	250	39·6	8·63	1 945	15·5	233	1896	„	3 blades. D. = 7·0′. P. = 7·10′. P. ratio = 1·01. A. = 18·7. A. ratio = ·486. Revs. = 280. App. slip per cent. = 21·0. (Durand.)
„ „ Concord .	1 723	228	36·0	14·1	3 314	17·0	213	..	„	3 blades. D. = 10·5′. P. = 13·2′. P. ratio = 1·26. A. = 26·5. A. ratio = ·306. Revs. = 152·4. App. slip per cent. = 14·4. (Durand.)
„ „ Bennington.	1 706	228	36·0	14·0	3 322	17·5	229	..	„	3 blades. D. = 10·5′. P. = 13·70′. P. ratio = 1·305. A. = 24·3. A. ratio = ·281. Revs. = 151. App. slip per cent. = 14·25. (Durand.)
British 3rd class cruiser Fearless	1 560	220	34·0	14·0	3 114	16·91	209	..	„	"Fearless." Propellers, D. = 10·5′. P. = 12·62′. P. ratio = 1·2. Exp. area = 24. A. ratio = ·278. Revs. 150·4. App. slip per cent. = 9·7.
Curzon, Elgin and Hardinge	800	250·1	38·1	6·0	*2 200	17·48	209	1912	Parsons geared turbines	Propellers outward turning. D. = 5′. P. = 4′ 8″. Revs., turbines 3 500, propellers 500.
Japanese light cruiser Tsukuski	1 350	210	31·9	14·5	2 400	16·0	208	1883	Recip. steam	Propellers, D. = 13′ P. = 16′. Revs. = 100.

* S.H.P.

TWIN-SCREW VESSELS 200 TO 300 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Block coefficient.	Midship-area coefficient.	Prismatic coefficient.
U.S. gunboat Yorktown . . .	·518	·867	·597
,, ,, Helena . . .	·549	·957	·574
,, ,, Concord . . .	·521	·850	·618
,, ,, Bennington . .	·520	·850	·612
British 3rd class cruiser Fearless .	·521	·920	·566

ACTUAL TWIN-SCREW VESSELS 200 TO 300 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	Type of engines.	
U.S. 3rd class cruiser Marblehead	2 054	257	37·0	14·4	5 303	18·44	191	..	Recip. steam	3 blades. Dia. = 11′. Pitch = 12′. Proj. area = 27·1. (Dyson.) Pitch ratio = 1·09. Area = 33·3. Area ratio = ·351. Revs. = 176·2. App. slip per cent. = 11·6. (Durand.)
U.S. 3rd class cruiser Montgomery	2 091	257	37·0	14·0	5 484	19·06	206	..	,,	3 blades. D. = 11′. P. = 12·75′. P. ratio = 1·16. Area = 29. A. ratio = ·305. Revs. = 180·3. App. slip per cent. = 16. (Durand.) Proj. area = 21·83.
U.S. 3rd class cruiser Detroit	2 068	257	37·0	14·46	5 155	18·71	206	..	,,	3 blades. D. = 11·0′. P. = 13·0′. P. ratio = 1·18. A. ratio = ·305. Revs. = 170·1. (Durand.) Proj. area = 21·83. (Dyson.)
Japanese protected cruiser Akashi	2 762	295	41·8	15·75	7 600	19·5	192	1895	,,	Propellers, D. = 12′ 5⅝″. P. = 15′ 1″. Revs. = 150.
Twin-screw steamer*	2 000	250	34·45	14·7	3 730	16·1	170	1897	,,	
Channel steamer Frederica	1 545	265	35·0	11·79	5 553	19·48	178	1890	,,	
,, Alberta	1 534	270	35·5	10·81	5 350	19·9	196	1900	,,	
British despatch vessel Surprise	1 544	250	32·5	13·83	3 046	17·0	215	..	,,	Propellers, D. = 11·0′. P. = 14·75′. P. ratio = 1·34. Exp. area = 24. A. ratio = ·253. Revs. = 132·1. App. slip per cent. = 11·4.
British 3rd class cruiser Barham *	1 830	280	35·0	13·25	5 870	20·07	225	1899	,,	See progressive trials.
Channel steamers Normannia and Hantonia	..	290·3	36·1	13·92	†6 100	20·4	..	1911	..	Two sets geared turbines. Revs. chosen to allow of propeller about 8 ft. dia. See Prof. Bile's paper. Φ coef. = ·902. 19·7 knots = 5 000 S.H.P.; 19·5 knots = 4 750 S.H.P.
Channel steamer Arundel	1 310	277	34·0	9·5	5 600	21·0	198	1900	Recip. steam	

* See progressive trials. † S.H.P

Twin-Screw Vessels 200 to 300 feet in Length. (Particulars independent of size.)

Nationality and name.	Block coefficient.	Midship-area coefficient.	Prismatic coefficient.
U.S. Marblehead	·525	·871	·603
U.S. 3rd class cruiser Montgomery	·550	·909	·605
,, ,, ,, Detroit . .	·526	·875	·601
British despatch vessel Surprise .	·481	·872	·552

ACTUAL TWIN-SCREW VESSELS 200 TO 300 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam.	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	Type of engines.	
Yacht Narcissus . . .	782	245	27·5	..	†1 250	14·5	..	1905	Parsons turbines	550 revs. Trial speed given. Sea speed 2 knots less.
Japanese torpedo gun-vessel Chihaya	1 238	275	31·625	9·06	6 000	21·0	177	..	Recip. steam	Propellers, dia. = 9′ 0¼″. Pitch = 11′ 11¾″. Revs. = 220.
Japanese torpedo gun-vessel Tatsuta	850	240	27·5	9·5	5 000	21·0	166	1894	,,	Propellers, dia. = 8′ 6″. P. = 13′ 3″. Revs. = 240.
U.S. Vesuvius	771	246	26·42	9·51	3 712	21·42	222	..	,,	Propellers, 3-bladed. D. = 7·75′. P. = 9·37′. P. ratio = 1·21. Area = 15·9. Area ratio = ·337. Revs. = 268·9. App. slip per cent. = 13·8.
British T.B.D. Lysander .	965	260	27·8	9·5	24 500	29·0	..	1913	Direct turbines	
,, ,, Badger . .	780	240	25·1	8·4	16 500	27	..	..	Geared turbines	
French T.B.D.'s Fourché and Faulx	725	246 w.l.	24·75	7·42	*18 500	33·2	..	1913	Direct turbines	680 revs. 8·19 knots per ton fuel burnt. Propellers, D. = 6′ 11″. P. = 6′ 5″. At 14·3 knots, 242 revs. 15·38 knots per ton fuel burnt. Oil fuel.
	725	237·5 b.p.	24·75	7·42						
British T.B.D. Lurcher . .	860	255	25·7	8·6	20 000	35·3	..	1911	,,	
Japanese T.B.D. Shirakumo .	372	216	20·75	6·82	7 600	31·0	203	1901	Recip. steam	Propellers, D. = 6′ 10¾″. P. = 9′ 1″. Revs. = 390.
U.S. T.B.D. Preble . . .	474·7	244 w.l.	23·5	6·5	7 110	27·55	179	1901	,,	206 tons weight of machinery.
Argentine T.B.D. Jujuy .	995	286·5 w.l. 280 b.p.	26·25 on w.l.	8·71	‡24 000	33	..	1912	Curtis A.E.G. turbines	640 revs. 4-bladed propellers. D. = 7·5′. Shaft centres = 10′ 6″ apart. Extreme beam = 27′.
German submarines U 21–32 (on surface)	650	213·18	20	11·83	*1 800	16	..	1911–12	Oil engines	
German submarine (on surface)	738	214·146	20	..	*4 000	20	..	1914	,,	
Japanese T.B.D. Akatsuki .	363	220	20·5	5·75	6 420	31	235	1902	Recip. steam	Propellers, dia. = 7′. P. = 9′. Revs. = 390.
U.S. T.B.D. Macdonough .	410	242·25	22·25	6·54	8 400	28·03	..	1901		

* B.H.P. † Equivalent. ‡ S.H.P.

TWIN-SCREW VESSELS 200 TO 300 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Block coefficient.	Midship-area coefficient.	Prismatic coefficient.	$\frac{V}{\sqrt{L}}$
Yacht Narcissus	53·2	..	..	..	·928
Japanese torpedo gun-vessel Chihaya	59·6	·549	..	..	1·268
,, ,, ,, Tatsuta	61·5	·475	..	..	1·257
U.S. Vesuvius	51·9	·436	·764	·571	1·367
British T.B.D. Lysander	55	·492	..	..	1·80
,, ,, Liffey	54·9	·424	..	..	1·713
,, ,, Badger	56·5	·539	..	..	1·743
U.S. T.B.D. Preble	32·7	·400	·770	·52	1·764
Argentine T.B.D. Jujuy	42·4	·531	..	..	1·949
	45·4	·544	..	..	1·972
German submarines U 21–32 (on surface)	67·2	·451	..	..	1·096
German submarine (on surface)	75·2	..	..	..	1·368
Japanese T.B.D. Akatsuki	34·1	·49	..	..	2·09
U.S. T.B.D. Macdonough	28·7	·407	·755	·54	1·802

Actual Single-Screw Vessels under 100 feet in Length.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.*	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	
Harbour ferry-boat	34·3	36·6	14	3·38	55	6·5	53	1904	One set comp. steam engines $\frac{7''-13''}{9''}\times 220$ revs., 1 propeller at each end of boat. Dia. = 8′. Pitch = 4′. Area ratio = ·56. 3 blades.
Motordrifter Pioneer II.	89	66·58 w.l.	18·57	7·33	30 †	5·5	110	1910	Int. comb. engine. Trim by stern $9\frac{1}{2}''$ in 12 ft.
Motor lifeboats, Royal National	13·25	38·33 w.l.	10·34 w.l.	3·166	40 †	..	..	1910	Propeller 22″ dia. Iron keel 2 tons. Breadth, exclusive of belting, 10·5′.
Hydraulically propelled steam lifeboat President Van Heel	30	53 w.l.	13·5	3·1	220	8·5	27	1895	About 3″ deeper aft of midship section than forward of it (stepped). See *International Marine Engineering*, Dec. 1907.
Scottish motor drifter	86	72	18·15	7·33	80 †	8·4	144	1910	Keel inclined aft $9\frac{1}{2}''$ in 12 ft. 500 revs. 5″ keel. *The Shipbuilder*, vol. iv, 1910.
Motor cutter	3·42	27	6·81	2·25	11·5	6·95	66·5	1903	10 B.H.P. See *Trans. Inst. Engineers and Shipbuilders, Scot.*, 1904.
Steam cutter	4·29	27	6·75	2·5	15	7·81	84	1903	*Ibid.* 13 B.H.P.
North Sea trawler	256	92 b.p.	21·7	9·208	230	9·0	124·5	1913	Propeller, D. = 9′.
Iwana	198	92·5	20·95	8·16	349	11·58	152	..	Propeller, 4 blades. D. = 7·5′. P. = 12·5′. P. ratio = 1·67. A. = 22·5. A. ratio = ·509. Revs. = 115·5. App. slip % = 18·8. (Durand.)
Tug Narkeeta	190	92·5	20·95	7·92	356	11·22	131	..	Propeller, 4 blades. D. = 7·5′. P. = 12·5′. A. = 22·5. Revs. = 111·8. App. slip % = 18·6. (Durand.) P. ratio = 1·07. A. ratio = ·509. App. slip % = 18·6.
Wahneta	176·5	92·5	20·95	7·6	378	11·63	130	..	Propeller, 4 blades. D. = 7·5′. P. = 12·5′. A. = 22·5. P. ratio = 1·67. Revs. = 114·6. A. ratio = ·509. App. slip % = 17·7. (Durand.)

* The mean draughts are given as far as possible ex keel. † B.H.P.

SINGLE-SCREW VESSELS UNDER 100 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	Displacement $\left(\frac{L}{100}\right)^3$.	Block coef.	Midship section coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
Harbour ferry-boat	38·23	2·615	4·14	700	·693	·95	·73	1·075
Motor drifter Pioneer II.	27·79	3·58	2·532	302	·344	(Fine)	..	·674
Motor lifeboats, Royal National	27·0	3·705	3·27	235	·37	·790	·47	
Hydraulically propelled steam lifeboat President Van Heel	25·7	3·9	4·39	202	·47	..	..	1·168
Scottish motor drifter	25·2	3·97	2·48	230	·314	(Fine)	..	·99
Motor cutter	25·22	3·96	3·128	173·5	·29	..	..	1·34
Steam cutter	25·0	4·0	2·7	218	·329	..	..	1·503
North Sea trawler	23·6	4·24	2·356	329·5	·486	·825	·59	·939
Iwana	22·66	4·41	2·568	255	·438	·750	·584	1·207
Narkeeta	22·66	4·41	2·645	244	·433	·742	·583	1·17
Wahneta	22·66	4·41	2·756	227	·419	·731	·578	1·214

ACTUAL SINGLE-SCREW VESSELS UNDER 100 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam.	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{2/3}V^3}{\text{power}}$	Date	
Launch No. 4	23·3	54·8	11·88	8·15	41·5	8·5	121	..	Mid.-area coef. = ·670. Prism. = ·596. Propeller, 3 blades. Dia. = 4·33′. Pitch = 7·0′. Revs. = 151·1. Pitch ratio = 1·62. Area ratio = ·464. Draught with keel = 3·88. (Durand.)
Barge Footah	95	65	14	5·0	100	7·2	78	1899	
Dutch tugboat	69	72	14·75	5·6	260	11·01	86·3	1897	
King Edward VII's launch	3·8 (about)	32	5·66	1·5	23	8·66	69	1908	20 B.H.P. Oil motor, 4 cyls. 4″×4½″. Revs. = 900. Weight of engine, batteries, and petrol for 60 miles = 5 cwt.
U.S. Inca	120	96·5	16·25	7·0	400	14	167		
Chili, R. East Cowes	15	60	9·5	2·97	270	19	154	1892	
Thornycroft torpedo launch	4·5	40	6·18	1·5	138	18	115	..	120 B.H.P. 4-cyl. oil engine 8″×8″. 900 revs. Weight of engine = 25 cwt.
Lookout	42·9	96	13·6	2·48	84·5	9·53	126	..	Midship-area coef. = ·750. Propeller, 4 blades. D. = 4·65′. P. = 8·4′. P. ratio = 1·81. A. = 7·94. A. ratio = ·468. Revs. = 152·3. App. slip per cent. = 24·4. Draught with keel = 3·7. With larger propeller a better result was obtained. D. = 5·0′. P. = 9·0′. A. = 9·34. I.H.P. = 95. Revs. = 140. 10·19 knots. 18 per cent. app. slip. (Durand.)
Clara	38·4	91	11·75	3·23	142	13·0	176	..	4 blades. D. = 3·88′. P. = 6·50′. A. = 5·03. A. ratio = ·425. Revs. = 252. App. slip per cent. = 19·6. (Durand). Midship-area coef. = ·658.
U.S. 3rd class T.B. Mackenzie	65	99·25	12·75	4·25	850	19·91	150		
Motor boat Napier I.	1·8	39·9	5·0	·63 (hull)	78	18·88	127	1905	Mid.-area coef. = 92. Wetted surface = 150. 66 M.P. or about 68 B.H.P. Revs. = 800. Total draught = 2·7.
,, Legra Hotchkiss	2·26	39·9	5·0	1·1	195	29·68	230	1905	Mid.-area coef. = ·663. 1·1 = draught of hull proper. The total draught is 2·17. B.H.P. = 170.

SINGLE-SCREW VESSELS UNDER 100 FEET IN LENGTH.

Nationality and name.	Block coefficient.	Midship-section coefficient.	Prismatic coefficient.
Motor boat Napier No. I. . .	·50	·92	·546
,, Legra-Hotchkiss . .	·362	·663	·544

ACTUAL SINGLE-SCREW VESSELS UNDER 100 FEET IN LENGTH.

Nationality and Name.	Tons displacement.	Length B.P.	Beam.	Mean draught.	Block coef.	$\frac{D^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Knots.	I.H.P.	Date.
Tug Manati . . .	114	69	16·5	8·0	..	91·5	10·525	300	1907
North Sea trawler . .	256	92	21·66	9·208	·486	124·5	9	280	1913
Tug Pelorus . . .	213	92	20·5	7·875	..	..	..	..	1911

Actual Twin-Screw Vessels 100–200 feet in Length.

Nationality and name.	Tons displacement.	Length B.P.	Beam	Mean Draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	Type of engines	
Tug	540	115	26	10·5	1 170	11·5	75	1902	Recip. steam	Propellers, 9′ dia. Area ratio = ·51 (trial). 103 revs.
Yacht	189·3	101	21·2	7·14	272	10·31	134	1896	,,	See progressive trials.
Tugboat Sea Rover	410	120	24·38	10·0	750	12·0	127	1902	,,	Engines, 17″–28″ × 21″ stroke. 175 lbs. W.P. Revs. = 110. Propeller, 4 blades. D. = 6·5′.
Paducah	1 065	174	35·0	12·25	1 217	12·823	180·7	..	,,	3 blades. D. = 7·25′. P. = 6·833′. Proj. area ÷ disc. area = ·307. App. slip per cent. = 17·05. Revs. = 229·4. (Dyson.)
U.S. gunboat Wheeling	1 000	174·1	34	12·03	1 050	12·88	204	1897	,,	Draught with keel = 12·03. 3 blades. D. = 6·75′. P. = 7·25′. Pitch ratio = 1·07. Area = 17·0. Area ratio = ·475. Revs. = 231·4. App. slip per cent. = 22·25. (Durand.)
,, ,, Marietta	991	174·1	34	11·45	1 028	13·02	214	1897	,,	3 blades. D. = 6·75′. P. = 7·25′. P. ratio = 1·07. Area = 17·0. Revs. = 231·3. Slip per cent. = 21·55. Draught with keel = 11·96. (Durand.)
Steam yacht Revolution	200	140	17	7·0	1 800	18·0	..	1906	Curtis turbines	Mr Speakman's paper, *Trans. Inst. E. & S. Scot.*, 1906. 650 revs. Propellers, dia. = 4·5′.

TWIN-SCREW VESSELS 100 TO 200 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Block coefficient.	Midship-area coefficient.	Prismatic coefficient.	
Tug	·602			
Yacht	·435	·67	·65	
Tugboat Sea Rover	·492			
Paducah	·520	·860	..	Propulsive efficiency (E.H.P. ÷ I.H.P. per cent.) = 55·87 for bare hull, and 63·52 with all appendages. (Dyson.)
U.S. gunboat Wheeling	·512	·896	·571	Dyson quotes propulsive efficiency as 53·33 with bare hull, and 60·48 with all appendages. Mid.-area coef. = ·858. Block coef. = ·508.
,, ,, Marietta	·512	·896	·571	Dyson gives E.H.P. ÷ I.H.P. per cent. as 56·64 bare hull, and 64·23 with all appendages.

Actual Twin-Screw Vessels 300 to 400 feet in Length.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	I. H. P.	Knots	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{power}}$	Date	
French battleship Dévastation	10 704	312	69·6	24·2	8 320	15·17	204	1901	
U.S. battleship Texas	6 315	301	64·08	22·5	8 610	17·8	224		
British battleship Rodney	9 690	325	68·0	26·7	11 610	16·92	190	..	4 blades. Dia. = 15·5′. Pitch = 19·42′. Revs. = 107·2. Pitch ratio = 1·25. Area ratio = ·382. Area = 72. App. slip % = 17·6. (Durand.)
U.S. battleship Iowa	11 363	360	72·23	24·04	11 834	17·09	212	..	3 blades. D. = 16·5′. P. = 20·0′. P. ratio = 1·21. A. = 75·7. A. ratio = ·354. Revs. = 109·5. Slip % = 20·85. (Durand.)
,, ,, Oregon	10 250	348	69·25	24·0	10 891	16·79	205	..	3-bladed propellers. D. = 15′. P. = 15·6′. P. ratio = 1·04. Exp. area = 66. A. ratio = ·373. Revs. = 128·25. App. slip % = 15. (Durand.)
French battleship Magenta	10 600	330	65·6	27·25	11 045	16·02	180	1901	
U.S. battleship Indiana	10 225	348	69·25	23·87	9 498	15·55	186		3-bladed propellers. D. = 15·5′. P. = 16′. P. ratio = 1·03. Exp. area = 53·9. A. ratio = ·285. Revs. = 131. App. slip % = 24·9. (Durand.)
,, ,, Massachusetts	10 265	348	69·25	24·08	10 128	16·21	198		3-bladed propellers, same as "Indiana." Mean revs. = 132·7. App. slip % = 22·65.
French battleship Hoche	10 997	333	65·6	27·25	11 300	16·0	179	1900	
Russian battleship	15 000 normal	386·5 w. l.	76	26 mean	16 000	18·0	221	..	Designed power and speed.
U.S. battleship Kentucky	11 538	368 ,,	72·25	23·5	12 082	16·9	203		
,, ,, Alabama	11 734	368 ,,	72·0	24	11 200	17·0	225		
Japanese battleship Fuji	12 450	374	73	26·5	13 500	18·25	241	1896	Propellers, D. = 17′. P. = 18′. Revs. = 120.
British battleship Barfleur	10 500	360	70	25·5	13 163	18·5	230	1894	
,, ,, Impérieuse	7 645	315	61	25·0	10 184	17·21	..	..	4 blades. D. = 18·16′. P. = 22·06′. P. ratio = 1·22. A. = 87. A. ratio = ·346. Revs. = 88. App. slip % = 10·1. (Durand.)
,, ,, Majestic	14 900	390	75	27·5	12 000	17·5	271	1895	
Japanese b.s. Itsukushima	4 216	301	57·17	19·82	5 830	16·7	208	1891	
British battleship Canopus	12 950	390	74	26	13 500	18·25	248	1900	
Argentine b.s. General San Martin	6 882	328	61·81	23·3	8 285	18·071	258	1898	4-bladed propellers. D. = 16′. P. = 23′ 7″. Mean revs. = 93·87. 17% app. slip.
Argentine b.s. General Belgrano	7 282	328	61·81	23·3	13 000	20·1	235	1897	

(All on this page are warships with reciprocating steam engines.)

TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Pris-matic coef.	$\frac{V}{\sqrt{L}}$.
French battleship Dévastation	22·3	4·49	2·875	353	·713	..	..	·86
U.S. battleship Texas	21·3	4·7	2·847	232	·51	..	..	1·027
British battleship Rodney	20·9	4·785	2·545	282	·575	·859	·669	·94
U.S. battleship Iowa	20·08	4·99	3·004	243·2	·636	·945	·673	·90
,, ,, Oregon	19·89	5·04	2·885	243·3	·62	·923	·672	·90
French battleship Magenta	19·9	5·03	2·408	296	·629	..	..	·884
U.S. battleship Indiana	19·89	5·04	2·9	243	·622	·931	·668	·834
,, ,, Massachusetts	19·89	5·04	2·875	243·4	·619	·927	·668	·87
French battleship Hoche	19·7	5·08	2·408	298	·646	..	..	·879
Russian battleship	19·66	5·09	2·921	260	·688	..	..	·916
U.S. battleship Kentucky	19·61 w.l.	5·1	3·07	232	·643	..	..	·882
,, ,, Alabama	19·56 ,,	5·11	3·00	236	·644	..	..	·886
Japanese battleship Fuji	19·51	5·13	2·755	238	·602	..	..	·944
British battleship Barfleur	19·45	5·15	2·746	225	·574	..	..	·976
,, ,, Impérieuse	19·36	5·165	2·44	245	·557	·844	·660	·971
,, ,, Majestic	19·21	5·2	2·726	251·5	·65	..	..	·888
Japanese battleship Itsukushima	19·0	5·27	2·882	155	·434	..	..	·962
British battleship Canopus	18·99	5·27	2·845	218·5	·604	..	..	·924
Argentine battleship General San Martin	18·81	5·31	2·65	195	·51	..	..	·998
,, ,, General Belgrano	18·81	5·31	2·65	206	.54	..	..	1·11

Actual Twin-Screw Vessels 300 to 400 feet in Length.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	
U.S. battleship Maine	12 250	388	72·2	23·5	15 600	18·15	204	1902	Weight of machinery and water, 1 396 tons.
Austrian battleship Habsburg	8 340	354	65·66	23·33	15 000	19·0	188	1900	
" " Erherzog Friedrich	10 433	390·5 w.l.	72	24·5 mean	18 130	20·57	232	1906	14 100 I.H.P. = 19·01 knots.
Italian b.s. Carlo Alberto	6 396	324·75 p.p.	59	23	13 116	19·11	186	1898	8 821 I.H.P. = 17·7 knots.
Germany—Wörth	9 878	354	64·0	24·37	10 228	16·6	206	..	3-bladed propellers. Dia. = 16·13′. Pitch = 17·72′. Pitch ratio = 1·10. Area = 58·0. Area ratio = ·284. Revs. = 109·2. App. slip % = 13·1. (Durand.)
French b.s. Chares Martel	11 693	392·5	71·0	27·5	14 996	18·1	204	1897	
British cruiser Arrogant	5 750	320	57·5	21·0	10 290	19·6	235	1896	
French cruiser Charlemagne	11 260	380·75 w.l.	67 5	28 max.	15 294	18·1	195	1896	9 123 I.H.P. = 16·15 knots.
British cruiser Gibraltar	7 700	360	61·0	24·75	10 553	20·4	314	1891	3 blades, gunmetal. Revs. = 105½. Propellers, D. = 16′. P. = 23′ 3″. Epm. = 32 lbs. per sq. in.
U.S. cruiser New York	8 480	380	64·25	23·89	16 948	21·0	228	1891	3-bladed propellers. D. = 16′. P. = 21′. P. ratio = 1·31. Exp. area = 69·1. A. ratio = ·344. Revs. = 134·8. App. slip % = 24·8. (Durand.)
Holland—Heemskerck	5 130	315 w.l.	52·5	16·5 mean	6 600	16·7	208	1906	Design.
Austria—Sankt Georg	7 185	383·75 w.l.	61·75	21·25 mean	15 270	22	260	1906	13 095 I.H.P. = 21·3 knots.
U.S.S. San Francisco	4 088	310	49·15	18·75	9 581	19·52	198	..	Propellers, 3 blades. D. = 13·5′. P. = 18·75′. P. ratio = 1·39. Exp. area = 57·6. A. ratio = ·402. Revs. = 124·8. App. slip. % = 15·5.
U.S.S. Newark	3 980	310·8	49·17	18·27	8 582	19·0	201	..	3-bladed propellers. D. = 14·5′. P. = 18·97′. P. ratio = 1·31. A. = 52·8. A. ratio = ·320. Revs. = 126·95. App. slip. % = 20·1. (Durand.)

(All on this page are warships with reciprocating steam engines.)

TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
U.S. battleship Maine	18·61	5·38	3·07	210	·65	..	..	·923
Austrian battleship Habsburg	18·56	5·39	2·813	188	·539	..	..	1·013
" " Erherzog Friedrich	18·49	5·41	2·948	175	·527	..	..	1·044
Italian battleship Carlo Alberto	18·19	5·5	2 565	186·6	·507	..	..	1·065
Germany—Wörth	18·1	5·53	2·63	223	·626	·920	·68	·884
French battleship Charles Martel	18·09	5·53	2·582	193·5	·534	..	..	·914
British cruiser Arrogant	17·99	5·56	2·74	175·6	·52	..	..	1·099
French cruiser Charlemagne	17·72	5·64	2·41	204	·547	..	..	·93
British cruiser Gibraltar	16·95	5·9	2·46	165	·496	..	..	1·078
U.S. cruiser New York	16·9	5·91	2·69	154·9	·509	·880	·578	1·079
Holland—Heemskerck	16·67	6·0	3·18	164	·656	..	..	·904
Austria—Sankt Georg	16·1	6·21	2·905	127	·50	..	..	1·125
U.S.S. San Francisco	15·86	6·3	2·621	137·3	·50	·836	·599	1·109
U.S.S. Newark	15·87	6·3	2·69	138·7	499	·863	·578	1·08

ACTUAL TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{power}}$	Date	
British cruiser Challenger	5 880	355	56	21·25	12 500	21·0	241	1904	7 700 I.H.P. = 18·5 knots.
U.S. cruiser Olympia	5 586	340	53·0	20·73	16 850	21·69	190	1892	4 hours' trial. 3 blades. Dia. = 14·75′. Pitch = 19·0′. Pitch ratio = 1·29. Exp. area = 68·0. Area ratio = ·398. Revs. = 139·25. App. slip % = 16·95. (Durand.)
,, Philadelphia	4 325	315	48·57	19·21	8 533	19·68	236	..	3 blades. D. = 14·5′. P. = 20·39′. P. ratio = 1·41. Exp. area = 57·2. A. ratio = ·346. Revs. = 119·55. App. slip % = 18·2.
British cruiser Hyacinth	5 600	350	54·0	20·5	10 100	20·0	251	1898	3 blades. Immersion = 5′ 7¼″. D. = 13·125. P. = 13·25′. Revs. = 170.
U.S. cruiser Baltimore	4 392	315	48·5	19·52	8 678	19·57	232	..	3 blades. D. = 14·5′. P. = 20′. P. ratio = 1·38. Exp. area = 57·2. A. ratio = ·346. Revs. = 118·05. App. slip % = 16. (Durand.)
Japanese cruiser Naniwa	3 727	300	46·18	18·6	7 120	18·77	224	1885	Propellers, D. = 14′. P. = 18′ 6″. Revs. = 122.
Forth	3 584	300	46·0	17·62	6 160	18·18	229	..	3 blades. D. = 13′. P. = 17·5′. P. ratio = 1·35. A. = 47. A. ratio = ·354. Revs. = 122·6. App. slip % = 14·2. (Durand).
U.S. cruiser Charleston	3 557	300	46·16	17·85	6 316	18·2	222	..	3 blades. D. = 14′. P. = 17·6′. P. ratio = 1·26. A. = 54·8. A. ratio = ·356. Revs. = 114·7. App. slip % = 8·6. (Durand).
British (old) despatch vessel Iris	3 290	300	46·08	18·08	7 714	18·57	183	1878	Famous in propeller research. Third series of trials.
U.S. cruiser Chicago	4 543	315	48·25	19·0	4 606	15·33	214	..	4-bladed propellers. D. = 15·5′. P. = 24·59′. P. ratio = 1·59. A. = 77·9. A. ratio = ·418. Revs. = 70·4. App. slip % = 10·2. (Durand.)
Austria—Kaiser Karl VI.	6 250	367	56·0	20·33	12 800	20·0	212	1898	
Pleasure-steamer City of Lowell	2 445	319	48·0	12·81	4 347	19·27	299	1894	See progressive trials for propellers.
Dutch cruiser Koningin Wilhelmina	4 600	327	48·81	20·0	5 900	17·0	230	1892	
Italian cruiser Marco Polo	4 511	327	48·25	19·5	10 543	19·0	178	1894	
Great Britain—Pique	3 623	300	43·66	17 5	9 154	19·75	199		
Holland—Tromp	5 300	331	48	18·5 max.	6 000	16·5	228	1905	Design.

(All of the above with reciprocating steam engines.)

TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Pris-matic coef.	$\frac{V}{\sqrt{L}}$.
British cruiser Challenger	15·78	6·34	2·631	131·5	·487	..	..	1·115
U.S. cruiser Olympia	15·6	6·42	2·462	142	·523	·883	·592	1·176
" " Philadelphia	15·41	6·49	2·53	138·5	·515	·874	·589	1·11
British cruiser Hyacinth	15·42	6·49	2·635	130·7	·509	..	..	1·07
U.S. cruiser Baltimore	15·39	6·5	2·483	141	·516	·863	·598	1·104
Japanese cruiser Naniwa	15·39	6·5	2·48	138	·507	..	..	1·084
Forth	15·33	6·52	2·608	133	·516	·918	·562	1·052
U.S. cruiser Charleston	15·38	6·5	2·582	131·7	·504	·869	·580	1·051
British (old) despatch vessel Iri	15·37	6·51	2·55	122	·488	·889	·549	1·073
U.S. cruiser Chicago	15·3	6·54	2·54	145·4	·551	·868	·635	·865
Austria—Kaiser Karl VI.	15·25	6·55	2·75	126·5	·523	..	..	1·046
Pleasure-cruiser City of Lowell	15·05	6·65	3·742	75·4	·435	·761	·575	1·08
Dutch cruiser Koningin Wilhelmina	14·91	6·7	2·44	131·6	·504	..	..	·942
Italian cruiser Marco Polo	14·75	6·79	2·472	129·1	·514	..	..	1·05
Great Britain—Pique	14·55	6·88	2·493	134	·553	..	..	1·14
Holland—Tromp	14·5	6·9	2·592	146	·631	..	..	·908

ACTUAL TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	
Japanese cruiser Akitsushima	3 110	301	43·12	17·45	8 400	19·0	174	1892	Propellers, dia. = 13′ 5¼″. Pitch = 17′ 6″. Revs. = 130.
Turkey—Hamidieh	3 800	345	47·5	16·0	12 500	22·2	213	1904	
Mars	11 282	387	58·1	24·7	4 150	13·2	279	..	Propellers, dia. = 14·75′. Pitch = 14·25′. Proj. area ratio = ·347. Percentage for appendages = 3·2. (Dyson.)
Japanese cruiser Chiyoda	2 400	310	42·6	14·0	5 600	19·0	219	..	Propellers, dia. = 10′ 3″. Pitch = 11′. Revs. = 206.
Russian Imperial yacht Standart	5 255	370	50·66	20·0	12 000	21·5	250	1896	Propellers, dia. = 16′. Pitch = 27′.
Japanese cruiser Nütaka	3 366	324	44·0	16·18	9 400	20·0	191	1902	Propellers, dia. = 12′ 6″. Pitch = 13′ 6″. Revs. = 185.
British Royal yacht Victoria and Albert	4 700	380	50·0	18·0	11 000	20·0	204	1899	3 blades. Dia. = 18·25′. Pitch = 17·5′.
Japanese cruiser Suma	2 756	307	40·18	15·21	8 384	20·0	188	1896	Propellers, dia. = 12′ 3⅜″. Pitch = 15′ 0⅛″. Revs. = 170.
,, ,, Takasago	4 160	360	46·66	17·0	13 070	22·5	225	1897	Propellers, dia. = 13′ 9⅜″ Pitch = 16′ 6″. Natural draught. Revs. = 165.
Swedish cruiser Fylgia	4 100	377·25	48·75	16·0	12 440	22·5	235	1907	
Japanese cruiser Yoshino	4 180	360	46·5	17·0	15 750	23·0	200	1892	Propellers, dia. = 13′ 9″. Pitch = 17′. Revs. = 165.
,, ,, Otowa	3 000	321	41·25	15·75	10 000	21·0	192	1903	Propellers, dia. = 11′ 6″. Pitch = 18′. Revs. = 200.
Channel steamer	3 095	322	41·3	14·67	6 250	18·8	226	1906	3 blades. Pitch ratio = 1·224. Revs. = 157. App. slip % = 12. Area ratio = ·411.
Italian cruiser Piemonte	2 500	300	38	15·0	12 786	22·3	223	1888	7 050 I.H.P. = 20·41 knots.
Ciudad de Monte Video	2 610	350	44	10	*5 200	18·75	241	1915	Geared turbines.
Merchant steamer	5 150	348	44·1	16·4	3 290	14·0	249	1906	Propellers, 3 blades. P. ratio = 1·177. A. ratio = ·353. Revs. = 105·6. App. slip % = 9·65.
British Admiralty yacht Enchantress	3 190	320	40·0	15·0	6 500	18·0	194	1904	
Japanese cruiser Chitose	4 760	396	49·0	17·62	15 500	22·87	219	1898	Propellers, dia. = 18′. Pitch. = 17′ 6″. Revs. = 154.
British cruiser Pyramus	2 155	300	36·5	13·62	7 303	20·7	203		

* S H.P.

(All of the above with reciprocating steam engines *except* where noted.)

TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH. (Particulars independent of size).

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Block coef.	Mid-ship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
Japanese cruiser Akitsushima	14·34	6·98	2·471	114	·48	..	..	1·095
Turkey—Hamidieh	13·77	7·26	2·97	92·5	·507	..	..	1·197
Mars	13·75	7·26	2·15	196	·799	..	..	·672
Japanese cruiser Chiyoda	13·73	7·28	3·043	80·5	·455	..	..	1·08
Russian Imperial yacht Standart	13·7	7·3	2·531	104	·491	..	..	1·118
Japanese cruiser Niitaka	13·59	7·36	2·72	99	·512	..	..	1·11
British Royal yacht Victoria and Albert	13·15	7·6	2·779	85·8	·481	..	..	1·027
Japanese cruiser Suma	13·09	7·65	2·64	95·3	·515	..	..	1·141
,, ,, Takasago	12·97	7·71	2·745	89·3	·51	..	..	1·186
Swedish cruiser Fylgia	12·91	7·74	3·05	76·4	·488	..	..	1·157
Japanese cruiser Yoshino	12·9	7·75	2·735	89·5	·515	..	..	1·212
,, ,, Otowa	12·85	7·79	2·62	90·8	·504	..	..	1·17
Channel steamer	12·82	7·8	2·812	92·8	·556	..	..	1·048
Italian cruiser Piemonte	12·67	7·9	2·532	92·6	·512	..	..	1·18
Ciudad de Monte Video	12·58	7·95	4·4	61	·594	·95	·625	1·005
Merchant steamer	12·69	7·89	2·69	122·1	·712	about ·932	about ·768	·75
British Admiralty yacht Enchantress	12·5	8·0	2·667	97·4	·581	..	..	1·007
Japanese cruiser Chitose	12·39	8·09	2·78	76·8	·49	..	..	1·148
British cruiser Pyramus	12·17	8·23	2·679	79·9	·507	..	..	1·196

ACTUAL TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	
British cruiser Pegasus .	2 000	300	36·5	12·7	7 127	21·2	212	1899	
German cruiser Emden .	3 544	364	44·25	15·75	16 390	24·12	203	1908	
Channel steamer Duke of Connaught	2 210	315	38·0	11·75	5 800	20·1	238	1902	4 bronze blades. dia. = 10·5′. Pitch = 13 5′ starb. 14·0 Port. Exp. area = 39·2. Proj. area = 30. Exp. area ratio = ·452.
German Imperial yacht Hohenzollern	4 180	382	45·92	18·2	9 634	21·53	269		4-bladed propellers (Durand). Revs. = 107. Dia. = 14·76′. Pitch = 22·64′. Pitch ratio = 1·53. Area = 52. Area ratio = ·304. Slip % = 10.
Channel steamers :—									
Duke of York . .	1 800	310	37·0	12·0	4 771	19·06	215	1894	
Anglia . . .	2 340	330	39·0	11·46	7 173	21·2	234	1900	
Duke of Cornwall .	2 120	315	37·0	11·66	5 520	19·75	231	..	Contract 12′ 8″ mean draught. 4-bladed propellers, dia. = 10′ 6″.
Duchess of Devonshire	1 720	300	35·0	9·79	5 000	19·0	197	1897	
Duke of Clarence .	1 800	311	36·0	11·0	5 757	18·8	171	1891	
Connaught . .	2 185	360	41·5	13·0	9 143	24·15	259	1896	
Chelmsford . .	2 500	300	34·5	15·0	4 800	18·2	231	1893	
Ulster . . .	2 950	360	41·5	13·42	9 500	23·8	298	1896	
Calais Douvres . .									
Jap. cruiser Miyako .	1 771	315	34·45	14·0	6 046	20·0	193	1899	Propellers, dia. = 11′ 1⅛″. Pitch = 18′ 1⅛″. Revs. = 195.
,, ,, Yayeyama .	1 583	318	34·45	13·33	5 360	20·0	203	1889	Propellers, dia. = 10′ 6″. Pitch = 16′. Revs. = 154.
British cruiser Forward .	2 790	365	39·18	13·875	15 000	25·25	213	1905	6 runs on mile.

(All of the above with reciprocating steam engines.)

TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
British cruiser Pegasus	12·16	8·22	2·875	74	·505	..	..	1·224
German cruiser Emden	12·15	8·24	2·81	73·5	·49	..	..	1·268
Channel steamer Duke of Connaught	12·06	8·29	3·232	70·8	·550	..	..	1·133
German Imperial yacht Hohenzollern	12·01	8·33	2·52	75	·459	·863	·53	1·10
Channel steamers:—								
Duke of York	11·93	8·39	3·081	60·5	·458	..	..	1·083
Anglia	11·81	8·47	3·403	65·3	·555	..	..	1·169
Duke of Cornwall	11·74	8·52	3·175	67·8	·546	..	..	1·114
Duchess of Devonshire	11·67	8·58	3·579	68·7	·585	..	..	1·097
Duke of Clarence	11·58	8·64	3·271	60·0	·512	..	..	1·068
Connaught	11·51	8·68	3·191	46·9	·394	..	..	1·273
Chelmsford	11·5	8·7	2·3	92·5	·565	..	..	1·05
Ulster	11·51	8·68	3·09	63·3	·514	..	..	1·257
Calais Doures								
Japanese cruiser Miyako	10·92	9·15	2·46	56·7	·408	..	..	1·128
,, ,, Yayeyama	10·82	9·24	2·581	49·3	·381	..	..	1·124
British cruiser Forward	10·71	9·34	2·82	57·4	·492	..	..	1·322

ACTUAL TWIN-SCREW VESSELS 300 TO 400 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	H.P.	Knots.	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{power}}$	Date	Type of engines.	
British scout cruiser Patrol .	3 000	370	38·75	14	16 460 I.H.P.	25·569	211	1905	Recip. steam	
,, ,, ,, Diamond	3 000	360	40	14·5	10 066 I.H.P.	22·17	212	1905	,,	245 revs. at 10 000 I.H.P. 7 145 I.H.P. = 19·3 knots.
,, ,, ,, Sentinel .	2 858	360	40	14·08	17 488 I.H.P.	25·07	182	1905	,,	
,, ,, ,, Adventure	2 670	374	38·25	13·5 max.	15 850 I.H.P.	25·42	199	1904	,,	
U.S. T.B.D. M'Dougal . .	1 010	300	30	9·0	17 741 S.H.P.	31·5	176	..	Direct turbines	Wetted surface = 8 450 sq. ft. Particulars from Mr G. S. Baker's book.
					11 187 S.H.P.	27·38	184			
					6 488 S.H.P.	24·25	221			
					2 687 S.H.P.	19·36	270			
					1 290 S.H.P.	15·65	298			
					177 S.H.P.	8·1	300			

Twin-Screw Vessels 300 to 400 feet in Length. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Pris-matic coef.	$\frac{V}{\sqrt{L}}$.
British scout cruiser Patrol . . .	10·48	9·55	2·768	59·3	·524	...	..	1·33
,, ,, Diamond . .	11·11	9·0	2·76	64·3	·503	..	..	1·169
,, ,, Sentinel . . .	11·11	9·0	2·842	61·2	·493	..	..	1·321
,, ,, Adventure . .	10·22	9·79	2·833	51·1	·484	..	..	1·32
U.S. T.B.D. M'Dougal	10·0	10·0	3·333	37·4	·436	·735	·595	1·82

ACTUAL TRIPLE-SCREW VESSELS UNDER 300 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date.	Type of engines.	
Russian Imperial yacht Livadia	4 400	235	153	7·66	12 350	15·275	77·4	..	Recip. steam	
British India Co.—Lhassa .	2 170	275	44	..	*6 000	18·0	168	1905	Parsons turbines	
Yacht Emerald	900	198	28·6	..	*1 400	15	224	..	..	900 revs.
Channel steamer Cæsarea . .										
Mr Barbour's yacht Lorena .	1 400	253	33·25	13·0	*3 800	18·02	193	1903	..	Centre screw 4′ 8″ diam. 550 revs. Wing screws 4′ 0″ dia. 700 revs.
Turbinia II (on Lake Ontario) .	1 100	260	33	9·5	3 500	19·0	209	..	..	Propellers 4′ 1½″ dia.
Channel steamer Dieppe . .	1 360	280	34·66	9·25	6 500	21·75	194·5	1904	..	Propellers 5′ 3″ dia. Centre 610 revs. Wings 630 revs.
,, ,, Brighton .	1 200	280	34·0	9·0	6 000	21·5	187	..	..	Centre 480 revs. Wings 510 revs.
Pleasure steamer King Edward	650	250	30	6·0	*3 500	20·48	185	..	Parsons turbines	1 screw on centre shaft. 4′ 9″ dia. 505 revs. 2 screws each wing shaft 3′ 4″ dia. 750 revs.
,, ,, Queen Alexandra	900	270	32	6·5	4 400	21·43	208	..	,,	Centre screw 750 revs. Wings 1 090.
Italian torpedo crusier Partenope	834	230	27·0	12·08	4 157	19·0	146	1890	Recip. steam	
,, ,, ,, Tripoli	831	230	26·0	10·46	3 016	19·8	228	..	..	All screws dia. = 5·75′. Pitch 7·125. Expanded surface = 7·57 sq. ft. Revs. = 297. Slip per cent. = 5·25.
Normand torpedo boat . .	95	125	14	..	2 200	26·5	176	..	..	
Italian torpedo crusier Goito .	812	230	25·66	9·5	2 620	19·0	228	1887	..	1 900 I.H.P. = 17 knots. 3 600 I.H.P. = 20 knots.
Mr Vanderbilt's yacht Tarantula	145	152·5	15·25	5·0	2 200	25·36	201	..	..	Propellers, one each shaft, 3′ dia.

* Equivalent.

TRIPLE-SCREW VESSELS UNDER 300 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Midship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
Russian Imperial yacht Livadia	65·1	1·538	19·97	339·5	·559	..	..	·996
British India Co.—Lhassa	16·0	6·25	..	104·5	..	..	..	1·086
Yacht Emerald	14·44	6·93	..	116	..	..	..	1·066
Channel steamer Cæsarea								
Mr Barbour's yacht Lorena	13·12	7·61	2·557	86·6	·448	..	..	1·135
Turbinia II (on Lake Ontario)	12·69	7·88	3·475	62·7	·472	..	..	1·18
Channel steamer Dieppe	12·37	8·08	3·748	62	·530	..	..	1·30
,, ,, Brighton	12·13	8·24	3·78	54·7	·49	..	..	1·286
Pleasure steamer King Edward	12·0	8·33	5·0	41·6	·506	..	..	1·297
,, ,, Queen Alexandra	11·84	8·44	4·92	45·8	·561	..	..	1·306
Italian torpedo cruiser Partenope	11·73	8·52	2·238	68·6	·389	..	..	1·253
,, ,, ,, Tripoli	11·3	8·85	2·485	68·4	·465	..	..	1·306
Normand torpedo boat	11·2	..	..	48·65	..	..	..	2·37
Italian torpedo cruiser Goito	11·15	8·97	2·7	66·8	·508	..	..	1·253
Mr Vanderbilt's yacht Tarantula	10·0	10·0	3·28	40·9	·436	..	..	2·052

ACTUAL TRIPLE-SCREW VESSELS UNDER 300 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	H.P.	Knots	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{power}}$	Date.	Type of engines.	
British T.B.D. Eden .	570	220	23·5	8·25	7 500 equiv. I.H.P.	26·2	166		Parsons turbines	3 shafts, 2 screws on each shaft. 3·25′ dia. 940 revs.
British T.B. 31–32 . .	287	178·6	18·75	6·2	4 000	26·5	205	1908	,,	
,, ,, 11–12 . .	225	172	18	5·25	3 750	26·0	244	1907	,,	
Palmer T.B. 35–36 .	298	179	17·9	6·6	4 000	26·0	196	1909		
British T.B.D. :—										
Parramatta . .	700	245·75	24·5	7·8	9 500 S.H.P.	28·48	191	1910		
Maori . . .	1 035	280	27	8·8	15 500 ,,	33·0	236	1909	Turbines	
Tartar . . .	872	270	26	9·1	14 500 ,,	35·67	285	1908	,,	
Mohawk . . .	865	270	25	8·9	14 500 ,,	34·51	258	1908	,,	
Turbinia . .	45	100	9·0	3·0	2 000 equiv. I.H.P.	32	207	1894	Parsons turbines	3 shafts, 3 screws on each shaft. 1·5′ dia. 2 300 revs. 2′ pitch.
,, . .	445	110	9·0	5·0 under propeller	2 200 S.H.P.	35·5	255	1894	,,	Propellers as above. 2 200 revs. Weight of hull 15 tons. Machinery 22. Coal and water 7·5.
U.S. T.B.D. Smith .	700	289	26	8·0	10 362 ,,	28·35	174	1909	,,	
,, Reid . .	700	289	26	8·0	12 734 ,,	31·82	200	1909	,,	228 tons machiuery.

TRIPLE-SCREW VESSELS UNDER 300 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Midship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
British T.B.D. Eden	10·68	9·37	2·85	53·5	·468	..	..	1·766
British T.B. 31–32	10·5	9·53	3·26	50·5	·483	..	..	1·988
,, ,, 11–12	10·48	9·55	3·43	44·3	·484	..	..	1·982
Palmer T.B. 35–36	10·11	9·89	2·711	52	·493	..	..	1·942
British T.B.D.:—								
Parramatta	10·03	9·98	3·14	47·4	·522	..	..	1·818
Maori	9·65	10·37	3·07	47·2	·544	..	..	1·972
Tartar	9·64	10·39	2·86	44·4	·478 5	..	..	2·172
Mohawk	9·26	10·8	2·81	44·0	·504	..	..	2·10
Turbinia	9·0	11·11	3·0	45	·583	..	..	3·2
,,	8·19	12·22	1·8 under propeller	33·41	·315	..	..	3·38
U.S. T.B.D. Smith	9·0	11·11	3·25	29	·408	..	..	1·67
,, Reid	9·0	11·11	3·25	29	·408	..	..	1·876

Actual Triple-Screw Vessels 400 feet in Length and upwards.

Nationality and name.	Tons displacement.	Length.	Beam.	Mean draught.	H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	
German battleship Westfalen	18 900	451·7 p.p. 470 w.l.	89	27·5	27 104 I.H.P.	20·3	219	1908	
French battleship Justice	14 635	438·75 w.l.	79·5	27·5	18 548 ,,	19·43	238	1906	11 530 I.H.P. = 17·94 knots. "Vérité," 20 433 I.H.P. = 19·26 knots.
Russia—Pobeida . .	12 674	401	71·5	26	14 500 ,,	18	219		
Germany—Hanover .	13 040	410 w.l. 398·5 p.p.	73·75 (Brassey)	25·25	22 492 ,,	19·16	173	1907	12 153 I.H.P. = 16·9 knots. 17 768 I.H.P. = 18·7 knots. F. T. Jane gives beam as 72′.
,, Oldenburg .	21 000	546 w.l.	93·5	29·5	35 000 ,,	21·4	214		
French battleship Suffren	12 750	410 ,,	70	28·25	16 715 ,,	18	191		
,, ,, Iéna .	12 052	400	68·18	27·5	16 500 ,,	18·2	192		
Japan—Kawachi . .	20 800	500	84	28	26 500 S.H.P.	20·5	246	1912	Curtis turbines.
Germany—Bluecher .	15 500	489 w.l.	80·33	27 max.	43 886 I.H.P.	25·86	246	1908	
,, Scharnhorst .	11 420	449·75 ,,	70·75	24·5	26 987 ,,	22·7	221	1908	
France—Victor Hugo .	12 416	476 ,,	70·25	26·5	28 735 ,,	22·86	223	1907	
Russian cruiser Gromoboi	13 200	472·5 ,,	68	27	14 500 ,,	20·0	307	1899	
U.S. cruiser Columbia .	8 050	411·6 w.l. & p.p.	58·19	24	18 509 ,,	22·8	256	. .	Revs. per min. Starb. = 134. Port = 132·9. Centre = 127·7. Slip %. Wings = 19·5. Centre = 15·9. Wing screws, dia. = 15′. Exp. surf. = 53·7. Centre, dia. = 14′. Pitch = 21·5′. Exp. surf. = 53·28. Area ratio = ·304 wing. ·346 centre.
,, ,, ,, .	7 375	418	58·4	22·5	18 000 ,,	22·8	249	. .	(From Mr Baker's book.) Wetted surf. = 28 700.
,, ,, Minneapolis	7 375	412 w.l. & p.p.	58·19	22·54	20 862 ,,	23·07	223	. .	4 hours' trial. Scotch boilers. 3 blades. Wing screws, dia. = 15′. Pitch = 22′. Proj. area ratio = ·234. Centre screw, dia. = 14′. Pitch = 21·5′. Proj. area ratio = ·261. Mean app. slip = 18·8. Revs. of centre screw = 228·7.

(All of the above with reciprocating steam engines *except* where noted.)

TRIPLE-SCREW VESSELS 400 FEET IN LENGTH AND UPWARDS. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
German battleship Westfalen	19·7 p.p.	5·08	3·235	205	·599	..	..	·955
	18 92 w.l.	5·285	3·235	182·3	·565	..	..	·936
French battleship Justice	18·12	5·52	2·89	173	·535	..	..	·929
Russia—Pobeida	17·81	5·62	2·75	196·7	·594	..	..	·899
Germany—Hanover	18·00 w.l.	5·56 w.l.	2·92	189·3 w.l.	·575 w.l.	..	..	·946 w.l.
	18·51 p.p.	5·4 p.p.	..	206 p.p.	·615 p.p.	..	..	·962 p.p.
,, Oldenburg	17·11	5·845	3·168	129·1	·488	..	..	·915
French battleship Suffren	17·09	5·85	2·479	185	·550	..	..	·889
,, ,, Iéna	17·03	5·87	2·48	188·4	·562	..	..	·91
Japan—Kawachi	16·8	5·95	3·0	166·5	·619 7	..	..	·918
Germany—Bluecher	16·42	6·09	2·972	132·9	·512	..	..	1·171
,, Scharnhorst	15·73	6·35	2·888	125·9	·513	..	..	1·071
France—Victor Hugo	14·74	6·79	2·65	115·4	·491	..	..	1·049
Russian cruiser Gromoboi	14·39	6·95	2·52	125·2	·533	..	..	·92
U.S. cruiser Columbia	14·12	7·08	2·423	115·5	·491	..	..	1·125
,, ,, ,,	13·98	7·16	2·596	101	·47	·905	·52	1·116
U.S. cruiser Minneapolis	14·11	7·09	2·58	105·6	·480	·862	·558	1·138

ACTUAL TRIPLE-SCREW VESSELS 400 FEET IN LENGTH AND UPWARDS.

Nationality and name.	Tons displacement.	Length.	Beam.	Mean Draught.	H.P.	Knots.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date.	Type of engines.	
French cruiser Gloire . .	10 000	452·75 w.l.	63·5	26·5	21 400 I.H.P.	21·58	208	1900	Recip. steam	22 000 I.H.P. = 22·5 knots.
" " Dupetit Thouars	9 517	452·75 "	63·66	24·5	20 382 "	21·38	186	1905	"	
					10 977 "	17·92	236			
France—Ernst Renan . .	13 427	515 "	70·5	26·75	37 700 "	24·24	214	1909	"	22 560 I.H.P. at 21·3 knots.
" Dupleix . . .	7 700	426·5 "	58·5	24·5	17 715 "	20·9	201	1903	"	
					10 022 "	18·5	246			
Russia—Aurora . . .	6 630	413	55·75	21	11 610 "	20·0	242	1900		
France—Waldeck Rousseau	13 780	515 "	70·5	27	36 110 "	23·1	196	1910	"	136 revs. at 23 knots.
French cruiser Guichen .	8 277 (sheathed)	436·33 "	55	27	25 455 "	23·55	210	1898	"	
				max.	18 500 "	20·0 (24 hrs.)	177			
Russian cruiser Askold .	5 981	426	49·25	20·6	20 017 "	23·357	209	1902	"	
Japanese liner Katori-Maru	18 750	490	61	28	11 700 " total I.H.P. and S.H.P.	16·75	284	1914	Recip. steam wings L.P. turbine centre	Engines $\frac{27-42-66}{48}\times 200$ lbs. 6 boilers, 15′ 6″ × 11′ 9″. F.D.
Orient liner Otaki . .	11 716	465·4 p.p.	60·3	20·083	6 857 "	15·02	256	1908		
Liners Tenyo Maru and Chiyo Maru	18 220	550 b.p.	63·3	27·355	8 950 S.H.P.	15·08	264	1908	Parsons turbines	
Jap. liners Chiyo Maru and Tenyo Maru (on voyage)	16 900	550	63	25·625	18 300 "	18·25	219	1909	"	
Allan liner Victorian . .	13 000	520	60·4	27·5	12 000 "	18·5	291	1905	"	Propellers, D. = 8·25′. Revs. = 275. Mr Speakman's paper, Inst. E. and S. Scot., 1905.
" " Virginian . .	17 000	520·4	60·3	29·5	12 000 "	19·0	379	1905	"	280 revs.*
Olympic	52 250	852·5 b.p.	92·5	34·5	46 000 " total I.H.P. and S.H.P.	21·0 on service	282	1911	Combination of recip. and turbines	30 000 I.H.P. of recip. engines, 16 000 S.H.P. of turbines.
Heliopolis and Cairo . .	15 000	545	60·4	22·5	18 000 S.H.P.	20·75	302	1907	Parsons turbines	
Cunard liner Carmania .	30 918	650·4	72·2	33·29	21 000 "	18·5	297	1905	"	Mr Speakman's paper. Propellers D. = 14′. Revs. = 185.

* 400 tons saving in machinery weight as compared with triple-exp. recip. Coal consumpt. 1·4 lb. per equiv. I.H.P. Water consumpt. for the turbines = 14 lbs. per S.H.P. hour.

TRIPLE-SCREW VESSELS 400 FEET IN LENGTH AND UPWARDS. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
French cruiser Gloire	14·01	7·14	2·395	107·9	·460	..	..	1·014
„ „ Dupetit Thouars	14·03	7·13	2·595	102·8	·472	..	..	1·005
								·844
France—Ernst Renan	13·69	7 31	2·637	98·5	·485	..	..	1·069
„ Dupleix	13·7	7·3	2·388	99·4	·442	..	..	1·011
								·895
Russia—Aurora	13·5	7·405	2·658	94·2	·480	..	..	·985
France—Waldeck Rousseau	13·69	7·31	2·61	100·8	·492	..	..	1·019
French crusier Guichen	12·6	7·94	2·036	99·8	·447	..	..	1·127
								·957
Russian cruiser Askold	11·54	8·66	2·39	77·4	·483	..	..	1·132
Orient liner Otaki	12·93	7·74	3·00	116·7	·729	..	..	·696
Liners Tenyo Maru and Chiyo Maru	..	..	..	109·6	·67	..	..	·64
Japanese liners Chiyo Maru and Tenyo Maru (on voyage)	11·46	8·74	2·456	101·8	·665	..	..	·779
Allan liner Victorian	11·6	8·62	2·195	106·9	·61	..	..	·796
Olympic	10·84	9·23	2·68	84·5	·673	..	..	·719
Heliopolis and Cairo	11·08	9·04	2·681	92·8	·709	..	..	·889
Cunard liner Carmania	11·1	9·0	2·258	109·3	·700	..	..	·754
Yacht Mahroussah	10·5	9·54	2·68	48·5	..	..	..	·900

ACTUAL QUADRUPLE-SCREW VESSELS WITH FOUR SHAFTS.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{Power}}$	Date.	Type of engines.	
U.S. scout Chester .	3 673	420	46·96 on w.l.	16·5	26 100 equiv. I.H.P.	26·52	171	1908	Parsons turbines	On 4 hours' trial.
Cunard liner Aquitania	49 430	868·7	97	34	56 000 S.H.P.	23·5	313	1914	Parsons turbines in series	
Imperator . . .	56 000	883·6	98·3	35·5	62 000 ,,	22·5	269	1912	,,	4 propellers, 16′ 5″ dia. 185 revs. 4 condensers, each 16 146 sq. ft. 4′ 51″ centrifugal pumps.
Lutetia . . .	15 600	579	64·1	23	19 000 collective B.H.P.	20·5 on service.	284	1913	Inner shaft recip. steam engines. Outer, Parsons turbines	
French liner France .	26 760	689·2	75·6	29·83	47 000 S.H.P.	25	298	1912	Parsons turbines	
British cruisers Newcastle, Glasgow, Gloucester, Liverpool	4 800	430 b.p.	47	15·25	24 669 ,,	26·4	211	1910	,,	For curves of S.H.P., speed, and revs. of the "Town" class, see *The Shipbuilder*, vol. iv, No. 15, and vol. v, No. 18.
					23 000 ,,	26·25	223			
					14 051 ,,	23·34	258			
T.B.D. Viper . .	390	210	21	6·75	13 000 equiv. I.H.P.	36·58	201	1900	..	4 shafts. 2 screws on each shaft, 3′ 4″ dia. 1 180 revs. 10 300 I.H.P. = 33·83 knots. 8 350 I.H.P. = 31·118 knots. 750 I.H.P. = 15 knots.
,, Velox . .	440	210	21	7·25	7 000 ,,	27·1	164	1902	Turbines outer shafts. Recip. cruising engines on inner shafts	400 revs. 7½″ – 11″ – 16″ × 9″ stroke. Outer propellers, 4′ dia. 890 revs.
,, Swift . .	1 800	345	34·166	10·5	33 000 S.H.P.	36·0	192	1909	Parsons turbines	
,, Cobra . .	450	228	20·5	7·25	10 000 equiv. I.H.P.	34·5	240	1901	Turbines	4 shafts. 3 screws on each shaft. 2′ 9″ dia. 1 050 revs.

QUADRUPLE-SCREW VESSELS WITH FOUR SHAFTS. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
U.S. Scout Chester	11·17	8·95	2·826	49·6	·395 5	·73	·542	1·294
Cunard liner Aquitania	11·17	8·95	2·85	75·6	·604	..	..	·798
Imperator	11·11	8·99	2·768	81·4	·636	..	..	·756
Lutetia	11·09	9·04	2·786	80·4	·638	..	..	·854
French liner France	10·98	9·11	2·583	81·9	·608	·972	·62	·954
British cruisers Newcastle, Glasgow,	10·91	9·15	3·081	60·4	·545	..	..	1·274
Gloucester, Liverpool								1·267
								1·127
T.B.D. Viper	10·0	10·0	3·11	42·15	·459	..	..	2·522
,, Velox	10·0	10·0	2·9	47·5	·482	..	..	1·87
,, Swift	9·9	10·11	3·25	43·9	·509	..	..	1·938
,, Cobra	9·2	10·89	2·83	48·6	·474	..	..	2·38

ACTUAL TRIPLE-SCREW VESSELS 300 TO 400 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	H.P.	Knots	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{Power}}$	Date.	Type of engines.	
French battleship Henri IV.	8 948	350 w.l.	73	24·75 max.	11 500 I.H.P.	17·5	201	1900	Recip. steam	Design.
,, ,, Masséna.	11 924	380·5	66·25	27 ,,	13 500 ,,	18·0	226	1896	,,	Design.
Germany—Barbarossa .	10 790	384 w.l.	65·5	27 ,,	13 940 ,,	18·0	205	1908	,,	7 360 I.H.P. = 15·5 knots.
,, Wittelsbach .	11 830	400 w.l.	67	28 ,,	14 483 ,,	18·0	209	1912	,,	10 000 I.H.P. = 16.
,, Prinz Adalbert .	9 050	394 w.l.	65	25·75 max.	17 700 ,,	20·5	212	1903	,,	
Union Co., New Zealand—Loongana	2 400	300	43	12·5	equiv. 6 300 I.H.P.	20·2	235	1905	Parsons turbines	
Channel steamers:—										
Princess Maud . .	1 750	300	40	10·5	6 500 ,,	20·7	198	1904	,,	
Manxman . . .	2 000	330	43	10·5	8 500 ,,	23·14	281	1904	,,	
Londonderry . . .	1 950	330	42	10·5	7 000 ,,	22·3	247	1904	,,	
Duke of Cumberland .	..	330·7	41·1	13	..	21·0	..	1910	Turbines	Propellers, D. = 5′ 10″. Pitch = 5′ 3″. Revs. 502.
Ben My Chree . .	3 353	375	46·2	13·42	..	24·12	..	1908	Parsons turbines	Propellers all 7′ 2″ dia. × 6′ 8″ pitch. Revs. 456.
Chinese cruiser Ying Swei .	2 460	330	39·5 mld.	13	6 375 S.H.P.	21·21	274	1912	,,	556·5 revs. At 1 100 S.H.P. coal in lbs. per S.H.P. hour = 2·6. At 13·14 knots. S.H.P. = 1 278. Revs. = 330·6.
Channel steamer St George	2 740	352	41·1	13·08	10 000 ,,	23	239	1906	,,	H.S. = 17 865. G.S. = 462. 185 lbs. steam.
Jan Breydel	2 000 (about)	348	40	10 max.	..	24·288				
Channel steamer Princess Elizabeth	1 950	350	40	9·59	equiv. 11 000 I.H.P	24	179·6	1905	Turbines	490 revs. Astern 415 revs. = 16 knots.
British cruiser Amethyst .	3 000	360	40	14·5	14 000 S.H.P.	23·4	190·3	1908	Parsons turbines	

TRIPLE-SCREW VESSELS 300 TO 400 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
French battleship Henri IV.	20·85	4·8	2·95	209	·495	..	..	·935
,, ,, Masséna	17·4	5·75	2·451	217	·714	..	..	·923
Germany—Barbarossa	17·06	5·86	2·425	190·2	·556	..	..	·919
,, Wittelsbach	16·76	5·97	2·392	185	·552	..	..	·90
,, Prinz Adalbert	16·5	6·06	2·522	147·9	·481	..	..	1·035
Union Co., New Zealand—Loongana	14·38	6·98	3·44	88·9	·521	..	..	1·168
Channel steamers:—								
Princess Maud	13·33	7·5	3·81	64·9	..	..	..	..
Manxman	13·02	7·68	4·095	55·7	·47	..	..	1·273
Londonderry	12·71	7·86	4·00	54·4	·47	..	..	1·229
Duke of Cumberland	12·44	8·05	3·16	..	..	..	..	1·155
Ben My Chree	12·32	8·11	3·441	63·6	·505	·894	·565	1·246
Chinese cruiser Ying Swei	11·98	8·35	3·04	68·5	·508	..	..	1·169
Channel steamer St George	11·67	8·58	3·144	62·9	·507	..	..	1·228
Jan Breydel	11·5	8·7	4·00	47·5	·503	..	..	1·301
Channel steamer Princess Elizabeth	11·42	8·75	4·18	45·6	·509	..	..	1·283
British cruiser Amethyst	11·11	9·00	2·76	64·4	·503	..	..	1·234

ACTUAL QUADRUPLE-SCREW VESSELS WITH FOUR SHAFTS.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	
French battleship Danton .	18 028	476 w.l.	84	27	22 500 I.H.P.	20·18	250	1911	
U.S. battleship Wyoming .	26 000	554 ,,	93·25	28·5	31 437 S.H.P.	21·2	266	1912	
					20 784 ,,	19·21	300		
British battleships :—									
Collingwood . . .	19 250	500 b.p. 530 w.l.	84	27	26 319 ,,	21 5	271	1910	Weight of main and auxiliary engines = 1 072½ tons + water to working level = 1 983½. 1·8 lb. consumption per H.P.
Bellerophon . . .	18 600	490 p.p. 520 w.l.	82	27	24 100 ,,	21·9	306	.	
Neptune . . .	19 900	510 p.p. 540 w.l.	85	27	27 721 ,,	21·78	274	1911	Machinery with auxiliaries = 1 109 tons + water to working level = 2 036½.
					18 373	19·0	275		
Dreadnought . .	17 900	500 p.p. 520 w.l.	82	26	24 712 ,,	21·02	258	..	22 200 tons at 31′ max. draught.
					16 930	19·3	291		
Ajax and King George V.	23 000 abt.	555 p.p. 589 w.l.	89	27·5	28 005 ,,	22·13	313		
Germany — Moltke and Goeben	22 640	590·5 w.l.	96·75	27	85 700	28·2	210	1912	
					74 000	27·2	219		
British battleships Colossus and Hercules	20 000	540 w.l.	86	27	28 750 ,,	21·5	255		
					18 000	19·6	308		

(All of the above with Parsons turbines.)

QUADRUPLE-SCREW VESSELS WITH FOUR SHAFTS. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Mid-ship area coef.	Pris-matic coef.	$\frac{V}{\sqrt{L}}$.
French battleship Danton	17·8 w.l.	5·62	3·11	167·5	·584	.	..	·924
U.S. battleship Wyoming	16·82 „	5·94	3·27	153	·619	..	..	·900
								·815
British battleships :—								
Collingwood	16·8 p.p.	5·95	3·11	154	·594	..	..	·962 b.p.
	15·85 w.l.	6·31	..	129·5	·56	..	..	·934 w.l.
Bellerophon	16·73 p.p.	5·98	3·04	158·1	·600	..	..	·99 p.p.
	15·76 w.l.	6·345	..	132·3	·565	..	..	·96 w.l.
Neptune	16·67 p.p.	6·00	..	..	..	..	..	·964 p.p.
	15·74 w.l.	6·35	..	..	..	..	..	·817 w.l.
Dreadnought	16·4 p.p.	6 1	3·15	143·2	·588	..	..	·94 p.p.
	15·77 w.l.	6·345	..	127·4	·565	..	..	·922 w.l.
Ajax and King George V.	16·03 p.p.	6·24	3·24	134·7	·593	..	..	·939 p.p.
	15·11 w.l.	6·62	..	112·7	·559	..	..	·912 w.l.
Germany—Moltke and Goeben . . .	16·39 „	6·1	3·58	110·0	·514	..	..	1·159
								1·119
British battleships Colossus and Hercules	15·91 „	6·29	3·188	127	·559	..	..	·925
								·844

ACTUAL QUADRUPLE-SCREW VESSELS WITH FOUR SHAFTS.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	H.P.	Knots.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date.	
Britain—Orion, Conqueror, Thunderer	22 500 abt.	544·5 p.p. 577 w.l.	85	27·75	28 600 S.H.P. 18 000 ,,	21·5 19·0	277 304		
Germany—Von der Tann .	19 400 (or 18 700	558 ,, 561	85 87	27·5 26·5	79 802 ,, 71 500 ,,	27·4 27·6	186 208)	1910	
Britain—Indomitable .	17 250	530 p.p. 560 w.l.	78·5	26	46 000 ,,	26·75	278		
Germany—Seydlitz . .	24 640	656	93·5	27	100 000 ,,	29·2	211	1913	
Britain—Indefatigable and New Zealand	18 750	555 p.p. 578 w.l.	79·5	26·5	47 135 ,,	26·5	279		
Britain—Lion and Princess Royal	26 350	660 p.p. 675 w.l.	86·5	28 normal 30 max. 31·5 full load	70 000 ,,	28·0 designed 31·7 max. by patent log.	279		
Chinese cruiser Chao Ho .	2 750	330	42	13·25	6 000 ,,	22	349	1912	
German cruiser Lubeck .	3 200	341	43·25	16·5	14 000 ,, equiv. 10 000 I.H.P.	23·0 on trial 22 (design)	189 231	1905	Lubeck propellers.*—Wings: D. = 62·9″. P. = 56·4″. Pitch ratio = ·896. Proj. area = 12·9 sq. ft. $\frac{\text{Proj. area}}{\text{Disc area}}$ = ·6. Inner: D. = 68·8″. P. = 61·9″. Pitch ratio = ·898. Proj. area = 15·5 sq. ft. $\frac{\text{Proj. area}}{\text{Disc area}}$ = ·6. 22·55 knots. 13 879 S.H.P. by Föttinger torsionmeter. App. slip % = 25·79. 625 revs. Displacement on trial about 3 250.
Allan liner Alsatian . .	22 500	600 w.l. 570 b.p.	72	28·5	21 375 S.H.P.	20·0	300	1913	Trial on measured mile at Skelmorlie at this draught. 285 revs.
Cunard liner Lusitania, trial	37 080	762·2 b.p.	87·8	32·75	76 000 ,,	25·62	247	1907	
,, ,, Mauretania .	39 000	762·2 ,,	88	34	76 000 ,,	25·5	255	1907	
German cruiser Augsburg .	4 280	402	46	16·5	20 000 ,,	27	259	1909	
Britain — Weymouth, Falmouth, Dartmouth	5 250	430 p.p.	48·5	15·25	22 000 ,, 18 839 ,,	25·5 24·95	227 249		

* *International Marine Engineering*, 1908.

(All of the above with Parsons turbines.)

Quadruple-Screw Vessels with Four Shafts. (Particulars independent of size.)

Nationality and name.	Beam as percentage of length.	$\frac{\text{Length}}{\text{Beam}}$.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Midship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.
Britain—Orion, Conqueror, Thunderer .	15·61 p.p.	6·4	3·062	139·3	·613	..	..	·922 p.p.
	14·72 w.l.	6·79	3·062	117·1	·578	..	..	·895 w.l.
Germany—Von der Tann	15·22 w.l. (or	6·57	3·09	111·7	·52	..	..	1·16
	15·5	6·46	3·28	106	·506	..	..	1·165)
Britain—Indomitable	14·81 p.p.	6·75	3·02	116	·559 b.p.	..	..	1·161 p.p.
	14·01 w.l.	7·14	3·02	98·4	·529 w.l.	..	..	1·131 w.l.
Germany—Seydlitz	14·25	7·02	3·46	87·4	·52	..	..	1·14
Britain—Indefatigable and New Zealand	14·31 p.p.	6·99	3·00	109·9	·561 p.p.	..	..	1·125 p.p.
	13·76 w.l.	7·27	3·00	97·0	·539 w.l.	..	..	1·102 w.l.
,, Lion and Princess Royal . .	13·11 p.p.	7·63	3·09 normal	91·7	·576	..	..	1·089 p.p. (design) 1·077 w.l. (design)
	12·81 w.l.	7·8	3·09 normal	85·8	·564	..	..	1·234 p.p. (max. speed) 1·22 w.l. (max. speed)
Chinese cruiser Chao Ho	12·72	7·86	3·17	76·6	·524	..	..	1·21
German cruiser Lubeck	12·69	7·89	2·62	80·7	·46	..	..	1·19
Allan liner Alsatian	12·0 w.l.	8·33	2·525	104·1	·639	..	..	·816 w.l.
	12·63 b.p.	7·91	2·525	121·5	·673	..	..	·838 b.p.
Cunard liner Lusitania	11·51	8·69	2·68	83·6	·591	..	..	·929
,, ,, Mauretania	11·53	8·66	2·59	88	·598	..	..	·924
German cruiser Augsburg . . .	11·45	8·74	2·79	65·9	·49	..	..	1·346
Britain—Weymouth, Falmouth, Dartmouth	11·28 p.p.	8·87	3·18	66	·578	..	..	1·23 1·203

Particulars of other British warships are given on p. 342.

ACTUAL TWIN-SCREW VESSELS 500 TO 600 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	Type of engines.	
T.S.S. ——	14 800	513	60·7	23	6 500	15	314	1906	Recip. steam	78 revs. 100 tons coal per day.
St Louis	11 629	535	63·0	23·5	18 000	21·8	295	1895	,,	
Merchant steamer	15 400	500	58·26	25·5	9 440	16·55	297	1902	,,	3 blades. Pitch ratio = 1·25. Area ratio = ·32. Revs. = 91. Slip per cent. = 14·25.
Paul Lecat	15 100	510·7	61·6	24·33	10 500	17·245	305	1912	,,	Engines $\frac{30\frac{1}{2}'' - 43'' - 62\frac{1}{2}'' - 88\frac{1}{2}''}{53''} \times 215$ lbs. 12 cyl. boilers. Howden's F.D. Total H.S. = 29 000 sq. ft. Contract speed = 16·5 knots.
Fürst Bismarck	10 490	502	57·5	23·3	15 944	20·7	266	..	,,	3 blades. D. = 19·03′. P. = 27·89′. Pitch ratio = 1·47. Area = 86·1. Area ratio = ·303. Revs. = 90·8. App. slip per cent. = 17·1. (Durand.)
Smolensk	11 850	506	58	24·0	16 500	20·0	252	1901	,,	
Korea	18 400	550	63·2	27·0	17 900	20·0	312	..	,,	
Kenilworth Castle	14 180	570	64·6	20·794	12 500	19·0	321	1903	,,	
Saxonia (at sea)	25 100	580	64·2	31·83	9 950	15·5 at sea	320	1900	,,	*Trans. I.N.A.*, 1914. 76½ revs. 3 blades. Pitch ratio = 1·051. D. = 19·5′. P. = 20·5′.
La Provence	19 160	597	64·64	26·75	30 000	22·05	255	1906	,,	

TWIN-SCREW VESSELS 500 TO 600 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	Block coefficient.	Midship-area coefficient.	Prismatic Coefficient.
T.S.S. ——	·722		
St Louis	·515		
Merchant steamer	·726	·969	·74
Paul Lecat	·69		
Fürst Bismarck	·546	·897	·609
Smolensk	·59		
Korea	·69		
Kenilworth Castle	·647		
Saxonia (at sea)	·732		
La Provence	·649		

BRITISH WAR VESSELS BUILT DURING THE WAR 1914–1918.

Name.	Tons displacement.	Length B.P.	Length over all.	Beam.	Mean draught.	Block coef.	Knots.	S.H.P.	Machinery.	No. of screws.
Battleship Canada	28 000	625	661	92 ext.	28·5	·600	22·75	37 000	Turbines	4
Battle-cruiser Renown	26 500	750	794	90	25·5	·52	32	112 000	,,	4
Large light cruiser Furious	19 100	750	786	88	21·5	·471	nearly 31·5	90 000	Geared turbines	4
Light cruiser Raleigh	9 750	565	605	65	17·25	·539	30	60 000	,, ,,	4
T.B. flotilla leader, Scott class	1 800	320	332·5	31·75	10·5	·59	36	40 000 to 44 000	,, ,,	2
Monitor Erebus	8 000	380	405	88	11·0	·761	12	6 000	?	2
Patrol boat, "P" class	573	230	244·5	23·75	7·583	·484	22	4 000	Geared turbines	2
T.B.D., "R" and "S" classes	1 065	265	276	26·66	9·0	·586	36	27 000	,, ,,	2
T.B.D., "V" and "W" classes	1 300	300	312	29·5	9·0	·57	34	27 000	,, ,,	2
Single-screw sloop, Flower class	1 250	255·25	267·75	33·5	11·0	·465	17	2 400	Reciprocating	1
Twin-screw minesweeper	750	220	231	28	7·0	·609	16	1 800	,,	2
Submarine, "J" class	1 210 *1 820*	270	275	23	14·0	·487	19 *9·5*	3 600 *1 350*	Oil engines	3
,, "K" class	1 880 *2 650*	334	338	26·5	16·0	·489	24 *9*	10 000 *1 400*	Geared turbines	2
,, "L" class	890 *1 070*	222	231	23·5	13·5	·444	17·5 *10·5*	2 400 *1 600*	Oil engines	2

Submarine figures in italics are when *submerged*.

ACTUAL TWIN-SCREW VESSELS OVER 600 FEET IN LENGTH.

Nationality and name.	Coef.	Tons displacement.	Length.	Beam	Mean draught.	I.H.P.	Knots	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{power}}$	Date	Type of engines.	
Minnesota	..	33 000	608	73·5	33·0	10 000	14·0	282	1904	Recip. steam	Coeffts.: Block = ·790. Mid area = ·987 5. Prism. = ·80.
Franconia	..	24 290	60	71	29·5	12 349	16·53	308	1911	,,	*Trans. I.N.A.* (1914). Sea speed.
Laconia	..	24 290	60	71	29·5	11 776	15·6	271	1912	,,	,, ,, ,,
George Washington (at sea)	..	36 000 approx.	699·1 b.p.	78·2	33·25	20 500	18·75	350	1909	,,	Propellers, 3-bladed. Dia. = 21′ 4″. 83 revs. Engines $\frac{38''-57''-80''-112''}{67''}\times 213$ lbs. W.P. Independent air pumps. Coeffts.: Block = 6·94. Mid area. = ·96. Prism. = ·723.
Celtic	..	37 700	680·9	75·3	36·5	14 000	16·0	327	1901	,,	
Caronia	..	31 155	650	72·2	33·29	20 644	18·56	305	1905	,,	*Trans. I.N.A* (1914). Sea speed.
Cedric	..	32 000	681	75·25	32	16 000	16·5	283	1903	,,	
Campania	·643	18 000	601	65 2	25·0	30 000	22·75	270	1893	,,	
,,	..	21 628	601	65·2	29·66	27 650	21·75	..	1893	,,	At sea. *Trans. I.N.A.* (1914).
Minnetonka	·712	26 530	600·7	65·5	33·13	11 000	16·0	332	1902	,,	
Adriatic	..	40 790	709·2	75·5	30	16 000	16·0	305	1907	,,	
Kronprinzessin Cecilie	..	27 000	685·4	72·2	30	45 000	23·5	260	1907	,,	
Kaiser Wilhelm II.	..	26 500	684·3	72·3	29·5	40 000	23·5	288	1904	,,	
Kaiser Wilhelm der Grosse	..	20 880	626·7	66·0	28	30 000	22·79	299	1898	,,	
Deutschland	..	23 200	662	67·3	28·81	35 500	23·5	297	1904	,,	
Oceanic	..	28 500	685·7	68·3	32·5	27 000	20·7	307	1900	,,	

ACTUAL TWIN-SCREW VESSELS 500 TO 600 FEET IN LENGTH.

Nationality and name.	Tons displacement.	Length.	Beam.	Mean draught.	I.H.P.	Knots	$\Delta^{\frac{2}{3}}V^3$ power	Date	Type of engines.	
U.S. battleship Utah (4-screw)	21 825	510	88·25	28·5	* 28 477	21·6	277	1911	Parsons turbines	
U.S. b.s. Delaware	20 000	510 w.l.	85·25	27	29 025	21·5	252	1910	Recip. steam	3 blades. D. = 18·25′. P. = 19·75′. Proj. area ratio = ·328. App. slip % = 13·3. E.H.P. ÷ I.H.P. bare hull = ·55. 1·83 lbs. coal per I.H.P. hour for all purposes. 128 revs.
,, ,, North Dakota	20 000	510 ,,	85	27	* 31 400	21·6	238	1910	Curtis turbines	Revs. 263 for 21 knots; 229 for 19 knots; 142¼ for 12 knots. 3 blades. D. = 13′. P. = 10·33′. Proj. area ratio = ·482. App. slip % full speed = 24·33. E.H.P. ÷ I.H.P. = 45·36 bare hull percentage. 53·07 with appendages.
,, ,, Texas	27 000	573	95·25	28·5	* 28 100	21·1	301	1914	..	
Brazilian b.s. São Paulo	19 281	500	83	25	28 645	21·6	254	1910	Recip. steam	
U.S. b.s. Montana	14 500	502 w.l.	72·75	25	27 938	22·26	236	1908	..	3 blades. Revs. = 120·2. (Dyson) 22 knots at 25 800 I.H.P. D. = 18′. P. = 21·75′. Proj. area ratio = ·310. Slip = 14·9 %.
British cruiser Terrible	14 200	500	71·0	27·0	25 648	22·41	257	1896	Recip. steam	3 blades. D. = 19·5′. P. = 24′. Pitch ratio = 1·23. Area = 92. Area ratio = ·308. Revs. = 112·3. App. slip % = 15·7.
,, ,, Good Hope	14 100	500	71·0	26·12	31 088	23·05	228	1902	,,	3 blades. D. = 19′. P. = 22·5′. Proj. area ratio = ·216. App. slip. % = 18·6.
U.S. cruiser Colorado	13 670	502	69·5	23·92	25 000	22·24	252	1903	,,	
C.P.R. Missanabie (trial)	13 080	500 b.p. 520 w.l.	64 64	21·041	9 365	17·493	317	1914	,,	Propellers, D. = 17·0′. P. = 19·25′. Exp. area = ·95. Proj. area = 78. 4 blades. 103·22 revs. App. slip = 10·76 %.
(City of) Paris	11 550	521	63·2	21·25	14 590	20·0	281	1899	,,	See progressive trials.
Cunarder Transylvania	19 480	548·3	66·5	27·5	† 10 000 (approx.)	15·5	270	1914	Geared turbines	Designed speed at sea. Mr Peskett's paper, *I.N.A.*, 1914.
,, Tuscania	22 000	548·3	66·5	30·5 fully loaded	* 11 000	16·5	320	1914	,,	Turbines, 1 707 revs. Propellers, 137 revs. 11·25 lbs. steam per S.H.P. hour.
Orient liner Osterley	15 280	535	63·2	24·25	13 790	18·76	295	1909	Recip. steam	
Empress of Britain	19 600	550	65·8	27·5	18 750	19·78	299	1906	,,	
Cunarder Carpathia	12 245	540	64	31·375	7 885	13·9	297	1903	,,	*Trans. I.N.A.* (1914).

* S.H.P. † Approx. S.H.P.

TWIN-SCREW VESSELS 500 TO 600 FEET IN LENGTH. (Particulars independent of size.)

Nationality and name.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Block coef.	Midship area coef.	Prismatic coef.	$\frac{V}{\sqrt{L}}$.	
U.S. battleship Utah (4-screw)	164·8	·583 7	·979 2	·597	·956	Dyson gives E.H.P. ÷ I.H.P. percentage = 48·5 for bare hull, and 56·16 with appendages, taking S.H.P. = ·92 I.H.P.
,, ,, Delaware	151	·600	·978	·615	·953	Dyson gives propulsive efficiencies as above = 55 and 64·9.
,, ,, North Dakota	151	·599	·978	·614	·956	Dyson gives propulsive efficiency with bare hull = 45·36, and 53·07 with appendages.
,, ,, Texas	143·6	·608	..	..	·882	
Brazilian battleship São Paulo	154·2	·65	..	..	·966	
U.S. battleship Montana	114·7	·556	·950	·586	·994	Dyson gives E.H.P. ÷ I.H.P. percentage = 49, bare hull, and 56·64 with appendages.
British cruiser Terrible	118·6	·518	..	..	1·003	
,, ,, Good Hope	113	·532	..	..	1·031	
U.S. cruiser Colorado	108	·581	·972	·599	·993	
C.P.R. Missanabie	92·3	·649	·977	·665	·733	
(City of) Paris	83·6	·581	·956	·60	·68	
Cunarder Transylvania						
,, Tuscania	133·7	·692	..	..	·754	
Orient liner Osterley	99·7	·653	..	..	·811	
Empress of Britain	118	·69	..	..	·843	

Mr G. S. Baker's Models, Set B. Corrected for ships of 400 ft. b.p. Beam, 52·6 ft. Mean draught, 23·21. Midship area coef. = ·98. $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}25$. Prismatic coef. : entrance = ·57 ; run = ·584. $\frac{\text{Length}}{\text{Beam}} = 7{\cdot}6$.

Beam, 13·16% of length. Parallel body, 10·44% of length. $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ assumed = ·50 in calculating $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.

Model No.	Tons displacement.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Coefficients. Block.	Coefficients. Prismatic.	Ratio. $\frac{\text{Length entrance}}{\text{Length run}}$	$\frac{V}{\sqrt{L}}$ / Knots	·425 / 8·5	·541 / 10·81	·58 / 11·6	·618 / 12·36	·656 / 13·12	·695 / 13·9	·733 / 14·67	·773 / 15·46	·811 / 16·22	·85 / 17·0	·889 / 17·78	·927 / 18·54	·966 / 19·31	1·005 / 20·1
17B	8 465	132·4	·606	·619 5	·566	E.H.P.	..	868	..	1 356	..	1 896	2 215	2 740	3 475	4 555	6 105			
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	..	302	..	289	..	294	296	280	255	224	191			
17A	8 460	132·3	·606	·619 5	·684	E.H.P.	..	851	..	1 278	..	1 912	2 198	2 580	3 113	3 981	5 300	7 065	8 910	10 700
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	..	308	..	307	..	292	298	297	285	257	220	187	168	158
14B	8 450	132·0	·605	·617 6	·904	E.H.P.	..	863	1 067	1 300	1 585	1 963	2 350	2 682	3 108	3 770	4 754	6 100	7 790	9 250
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	..	304	303	301	295	284	278	285	285	270	245	217	192	182
16B	8 434	131·8	·603	·615	1·214	E.H.P.	..	856	1 059	1 281	1 552	1 860	2 272	2 851	3 375	3 850	4 655	5 890	7 440	9 100
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	..	305	306	306	301	300	288	269	263	265	250	225	201	185
16D	8 422	131·6	·603	·615	1·624	E.H.P.	..	853	..	1 291	..	1 860	2 264	2 848	3 373	3 840	4 670	5 886	7 410	9 110
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	..	306	..	302	..	298	288	268	262	265	249	224	201	184

The speeds given above have been calculated from Mr Baker's K values by Mr Froude's formula $V = \frac{\Delta^{\frac{1}{6}}}{{\cdot}5834} \times K$, and the E.H.P. also by Mr Froude's constant system, $\text{E.H.P.} = \frac{\Delta^{\frac{2}{3}}}{427{\cdot}1} \times C \times V^3$.

Suitable maximum service speeds in heavy type.

Professor Sadler's Models, Series F 7. *Transactions of the American Society of Naval Architects and Marine Engineers*, 1907. $\frac{\text{Length}}{\text{Beam}} = 8$. Beam, 12·5% of length. Lines representing transatlantic intermediate type of merchant ship. Service speed, $\frac{V}{\sqrt{L}}$ = ·60 to ·75. Limit of economical speed about ·725. For a speed $\frac{V}{\sqrt{L}}$ = ·60 to ·65 the form would be fuller than that of this series. The humps occur at speeds $\frac{V}{\sqrt{L}}$ = about ·45, ·60, and ·79.

	Fine ends.	Full ends.	$\frac{\text{Beam}}{\text{Draught}}$.	$\frac{\text{Length}}{\text{Draught}}$.	Coefficients. Block.	Coefficients. Prismatic.	Coefficients. Midship.	Longitudinal distribution of displacement.	Residuary resistance in lbs. per ton of displacement for various speeds $\frac{V}{\sqrt{L}}$: ·45	·50	·55	·6	·65	·70	·75	·80	·85	·90	·95
F 7 (5).								Fine bow, full stern	·83	1·0	1·25	1·48	1·74	2·025	2·57	4·76	6·26	8·58	13·95
	Fine ends.	Full ends.	3·0	24	·697	·734	·949	Fine bow, fine stern	1·1	1·125	1·38	1·32	2·1	2·4	3·6	5·24	6·3	8·63	15·2
Coefficient W.L.	·795	·807						Full bow, fine stern	·68	·94	1·35	1·875	2·7	3·81	5·5	8·27	11·0	12·71	
½ angle, mean . ½ angle W.L. .	10·8° 15·5°	19·0° 28·5°						Full bow, full stern	·75	1·0	1·4	2·062	3·0	4·27	6·1	8·5	11·74	14·13	
F 7 (6).								Fine bow, full stern	·75	·96	1·13	1·37	1·61	1·875	2·7	5·05	7·0	8·81	
	Fine ends.	Full ends.	2·5	20	·715	·747	·956	Fine bow, fine stern	1·0	1·125	1·13	1·5	1·775	2·13	3·373	5·38	6·576	9·0	
Coefficient W.L.	·830	·812						Full bow, fine stern	·575	·825	1·087	1·54	2·25	3·28	4·625	7·375	10·75		
½ angle, mean . ½ angle W.L. .	20·4° 16·4°	11·4° 30·5°						Full bow, full stern	·56	·82	1·175	1·73	2·51	3·67	5·22	7·375	10·875		
F 7 (7).								Fine bow, full stern	·75	·97	1·18	1·41	1·65	2·03	2·85	5·54	7·75		
	Fine ends.	Full ends.	2·143	17·143	·733	·760	·964	Fine bow, fine stern	·76	·875	1·188	1·525	1·874	2·2	3·225	5·45	7·15		
Coefficient W.L.	·852	·829						Full bow, fine stern	·50	·725	1·088	1·525	2·213	3·088	4·438	6·875			
½ angle, mean . ½ angle W.L. .	.. ..	.. ..						Full bow, full stern	·575	·82	1·18	1·75	2·52	3·5	4·76	6·97			

For the fore body it is advantageous to have a comparatively long parallel body and fine bow, while with the after body the best results "seem to be obtained by adopting a form with a more gradual diminution of area from the midship section aft." So far as the fore body is concerned, this happens to be a cheaper ship to build than one with a long entrance.

Professor Sadler's Models, Series F 8. *Transactions American Society of Naval Architects and Marine Engineers*, 1908. $\frac{\text{Length}}{\text{Beam}} = 8.$ Beam 12·5 per cent. of length. $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}142.$ $\frac{\text{Length}}{\text{Draught}} = 17{\cdot}142.$

Coefficient: block = ·855; prismatic = ·869; midship = ·984. The dimensions, displacement, and coefficients were kept constant, and the distribution of displacement modified by altering the curve of sectional areas.

Rough approximation of percentage of parallel body.		Longitudinal distribution of displacement.	Residuary resistance in lbs. per ton of displacement at various speeds $\frac{V}{\sqrt{L}}$.						
Forward.	Aft.		·4	·45	·5	·55	·6	·65	
60	60	Full bow, full stern	·85	1·1	1·26	1·6	2·5	3·48	Best *
60	68	,, ,, fine stern	1·2	1·48	1·9	2·5	3·37	4·45	
68	60	Fine bow, full stern	1·03	1·36	1·84	2·5	3·5	5·15	
68	68	,, ,, fine stern	1·25	1·75	2·25	3·2	4·7		Worst *

The above particulars are for maximum draught. The resistance curves for the other draughts at which Professor Sadler's models were tried, followed the same general form.

* For vessels of block coefficient finer than ·8, the "best" and "worst" would be reversed, for the reasons given by Professor Sadler.

Professor Sadler's model, F 8. Tried with fine bow and fine stern, sharp ends, straight or even hollow ends of curve of sectional areas. Enlarged to 400 ft. ship. Displacement, 11 400 tons. $\dfrac{\Delta}{\left(\dfrac{L}{100}\right)^3} = 178{\cdot}2$. Dimensions, $400 \times 50 \times 23{\cdot}33$ ft. mean draught. Estimated wetted surface, 34 000 sq. ft. Froude's Ⓜ $= 5{\cdot}43$. Taylor's wetted surface constant $= 16{\cdot}6$ for W.S. $=$ 35 400. Parallel body about 52 per cent.

Ⓚ	$\dfrac{V}{\sqrt{L}}$.	Knots.	Residuary resistance lbs. per ton of displacement given by Prof. Sadler.	Residuary resistance in lbs.	Residuary H.P. from Prof. Sadler's figures.	Skin H.P.	E.H.P.	$\dfrac{\Delta^{\frac{2}{3}}V^3}{I.H.P.}$.	Assumed propulsive efficiencies with reciprocating steam engines, single screw.
·86	·35	7	·92	10 490	225·5	233·5	459	158	·42
·983	·4	8	1·25	14 250	350	342	692	165	·44
1·106	·45	9	1·75	19 960	552	477	1 029	163	·454
1·229	·5	10	2·25	25 690	789	642	1 431	163	·463
1·29	·525	10·5	2·7	30 800	993	738	1 731	158	·467
1·351	·55	11	3·2	36 500	1 232	841	2 073	153	·47
1·412	·575	11·5	3·85	43 900	1 550	955	2 505	...	·473
1·474	·6	12	4·7	53 600	1 975	1 077	3 052	...	·474
1·536	·625	12·5	6·0	68 400	2 625	1 209	3 834	...	·475
1·57	·64	12·8	7·1	81 000	3 183	1 291	4 474	...	...

The above values of $\dfrac{\Delta^{\frac{2}{3}}V^3}{I.H.P.}$ show the unsuitability of fine ends in a ship of this fulness. The explanation given by Professor Sadler is that there is a rather abrupt shoulder, where the lines run into the middle body, causing a secondary bow wave as well as a marked hollow in the wave profile at the stern. The performance is materially improved by fining the bilge diagonal.

$$\text{Skin H.P.} = {\cdot}009\,10 \times \text{wetted surface} \times {\cdot}003\,070\,7 \times V^{2{\cdot}83}$$

$$Ⓚ = \frac{{\cdot}583\,4}{\Delta^{\frac{1}{6}}} \times V$$

$$\text{E.H.P.} = \text{Resistance lbs.} \times V \times {\cdot}003\,070\,7.$$

The results from this form are the opposite to those obtained for F 7, where the block coefficient is ·733; *i.e.* F 8, with block coefficient ·85, should not be given a long parallel body and fine ends, as the above poor results show. It is better to have shorter middle body and fuller ends. The average service speed would be $\frac{V}{\sqrt{L}}$ = ·50 to ·55.

Professor Sadler's model, F 8. With full bow and full stern, round lines. Enlarged to 400 ft. ship.

(K)	$\frac{V}{\sqrt{L}}$.	Knots.	Residuary resistance lbs. per ton displacement given by Prof. Sadler.	Residuary H.P.	Skin H.P.	E.H.P.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.	Residuary resistance lbs.
·86	·35	7	·70	171·6	233·6	405·2	...	7 980
·983	·4	8	·85	238	342	580	...	9 690
1·106	·45	9	1·1	346·5	477	823·5	...	12 530
1·229	·5	10	1·26	441	643	1 084	234	14 370
1·29	·525	10·5	1·4	515	738	1 253	234	15 970
1·351	·55	11·0	1·6	616	841	1 457	231	18 250
1·412	·575	11·5	2·0	805	955	1 760	219	22 810
1·474	·6	12	2·5	1 050	1 077	2 127	206	28 500
1·536	·625	12·5	3·02	1 322	1 209	2 531	195	34 430
1·597	·65	13	3·48	1 584	1 350	2 934	190	39 680
1·621	·66	13·2	3·65	1 688	1 410	3 098	188	41 600

Skin H.P. = ·009 10 × wetted surface × ·003 070 7 × $V^{2·83}$.

This is a much better form than the last. For 10 knots minimum resistance would be obtained with about 38 per cent. parallel body, and for 12 knots 31 per cent. The curve of cross-sectional areas here is round at the ends. The fore body water-line is also round. "In other words, easy buttocks at each end rather than full below and fine above" (Sadler). The forward end transverse sections should be round V'd rather than U'd. Vessels with long proportion of parallel body require a long run and usually round lines aft, the entrance being relatively short.

Professor Sadler's models, Series F 6. (*Transactions of the American Society of Naval Architects and Marine Engineers*, 1908.) Service speed $\frac{V}{\sqrt{L}}$ = ·75 to ·90. In Series F 6 (1) the longitudinal distribution of displacement was modified successively by altering the curve of sectional areas. With fine ends there was about 20 per cent. of parallel middle body, and with full ends (round lines) no parallel body.

	Length / Beam	Beam / Draught	Length / Draught	Coefficients. Block.	Coefficients. Prismatic.	Coefficients. Midship.	Longitudinal distribution of displacement.	Residuary resistance in lbs. per ton of displacement for various speeds $\frac{V}{\sqrt{L}}$. ·45	·50	·55	·60	·65	·70	·75	·80	·85	·90	·925	·95	·975
F. 6 (1)	8	2·142	17·142	·653 3	·677 8	·963 8	Fine bow, full stern	·68	·8	1·0	1·25	1·5	1·81	2·15	2·56	3·16	4·75	6·3	8·8	12·0
							Fine bow, fine stern	..	..	1·0	1·28	1·58	1·85	2·08	2·45	3·25	5·19	7·0	9·6	12·65
							Full bow, full stern	..	..	1·0	1·3	1·74	2·25	2·85	3·5	4·18	5·2	6·6	8·55	
							Full bow, fine stern	..	..	1·2	1·56	2·03	2·67	3·35	4·0	4·62	5·61	6·9	8·76	11·4
F. 6 (2)	7·272	2·358	17·142	·594	·677 8	·874	Fine bow, fine stern	..	..	1·0	1·24	1·49	1·76	2·07	2·43	3·08	4·76	6·4	8·8	11·62
F. 6 (3)	7·272	2·358	17·142	·594	·664	·895	Fine bow, fine stern	..	·70	..	1·16	1·4	1·69	2·05	2·4	3·0	4·2	5·3	7·25	10·2

In this series a form with fine water-line is best, not too full at the bilge diagonal forward. For the aft body the curve of sectional areas should taper gradually from the midship section, "somewhat full on the water-line, and with an easy bilge diagonal." F 6 (1). The minimum resistance would be obtained with about 10 per cent. parallel body. (With 18 per cent. parallel body the resistance would be about 3 per cent. greater.) The curve of cross-sectional areas would be slightly hollow forward, and the fore body water-line slightly hollow. The forward end transverse section would be U'd. (A finer vessel for still higher speeds should have no parallel body, the curve of cross-sectional areas would be hollow forward and aft, the forward sections V'd, and the fore body water-lines should be straight if the speed is above $\frac{V}{\sqrt{L}}$ = 1·2, but this fine form is not included in the above table.)

Professor Sadler's model, F 6 (1), with fine bow and full stern. Enlarged to 400 ft. ship. Dimensions, 400 × 50 × 23·33 ft. mean draught. Displacement, 8 710 tons. Estimated wetted surface = 30 000 sq. ft. Froude's (M) = 5·94. Taylor's wetted surface constant = 16·3 gives 30 400.

$\frac{V}{\sqrt{L}}$.	Knots.	Residuary resistance in lbs. per ton of displacement.	Residuary resistance in lbs.	Residuary H.P.	Skin H.P.	E.H.P.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.
·45	9	·68	5 930	164	421	585	...
·5	10	·8	6 960	214	567	781	270
·55	11	1·0	8 710	294	742	1 036	272
·6	12	1·25	10 890	401·5	949	1 350	270
·65	13	1·5	13 070	521	1 190	1 711	271
·7	14	1·81	15 770	678	1 468	2 146	270
·75	15	2·15	18 750	864	1 785	2 649	270
·8	16	2·56	22 300	1 096	2 141	3 237	268
·85	17	3·16	27 540	1 438	2 542	3 980	261
·9	18	4·75	41 400	2 290	2 990	5 280	234
·925	18½	6·3	54 900	3 120	3 231	6 351	211
·95	19	8·8	76 700	4 476	3 483	7 959	182

Professor Sadler's models, F 6 (2) and F 6 (3), on the same basis of curve of sectional areas as F 6 (1), but with greater beam, are interesting.

F 6 (2). $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}358.$ $\frac{\text{Length}}{\text{Draught}} = 17{\cdot}142.$ $\frac{\text{Length}}{\text{Beam}} =$ 7·272. Beam, 13·76 per cent. of length. Coefficients: block = ·594; prismatic = ·677 8; midship = ·874.

Model F 6 (3) has the same increased beam as F 6 (2), the block coefficient is kept the same as in F 6 (2), but the prismatic coefficient is reduced to ·664. The end lines are the same for all three models F 6. The middle body is reduced in length by increasing the beam. These modifications give a more easily driven ship than F 6 (1).

F 6 (3). Length, 400 ft. Beam, 55·05 ft. Mean draught, 23·33 ft. Δ = 8 710. Take wetted surface = 29 000 sq. ft. Fine bow with fine stern.

$\frac{V}{\sqrt{L}}$.	Knots.	Residuary resistance in lbs. per ton of displacement.	Residuary resistance in lbs.	Residuary H.P.	Skin H.P.	E.H.P.	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.
·5	10	·7	6 100	187	549	736	287
·6	12	1·16	10 100	372	917	1 289	283
·65	13	1·4	12 200	487	1 150	1 637	283
·7	14	1·69	14 710	633	1 419	2 052	283
·75	15	2·05	17 880	823	1 725	2 548	280
·8	16	2·4	20 900	1 027	2 070	3 097	280
·85	17	3·0	26 130	1 364	2 460	3 824	272
·9	18	4·2	36 600	2 022	2 890	4 912	251
·925	18·5	5·3	46 150	2 623	3 122	5 745	233
·95	19	7·25	63 100	3 680	3 368	7 048	206
·975	19·5	10·2	88 900	5 330	3 626	8 956	175
1·0	20	12·61	110 000	6 710	3 892	10 602	160

Mr G. S. Baker's models, 1913, Set B.—*continued.*

$L = \frac{V}{\sqrt{L}} \times 1{\cdot}055\,2$. $S = \frac{S}{\Delta^{\frac{2}{3}}} \times {\cdot}093\,46$. Residuary resistance in lbs. $= \frac{\text{Residuary H.P.}}{V \times {\cdot}003\,070\,7}$.

Model No.	Estimated wetted surface. sq. ft.	Froude's S.	Results at various speeds.														
			$\frac{V}{\sqrt{L}}$	·425	·541	·58	·618	·656	·695	·733	·773	·811	·85	·889	·927	·966	1·005
			L	·439	·571	·612	·652	·693	·734	·774	·816	·856	·898	·939	·979	1·02	1·061
17B	30 400	6·84	Ⓒ	...	·706	...	·739	...	·726	·722	·763	·838	·954	1·118			
			$OSL^{-{\cdot}175}$	...	·56	...	·546	...	·534 8	·529 5	·525	·520 5	·516	·514			
			Skin H.P.	...	689	...	1 001	...	1 395	1 622	1 888	2 159	2 469	2 810			
			Resid. H.P.	...	179	...	355	...	501	593	852	1 316	2 086	3 295			
			Resid. resistance lbs. per ton Δ	...	·636	...	1·107	...	1·388	1·557	2·121	3·121	4·72	7·13			
17A	30 380	6·83	Ⓒ	...	·693	...	·696	...	·732	·716	·718	·75	·832	·97	1·118	1·272	1·356
			$OSL^{-{\cdot}175}$	...	·559 4	...	·545	...	·534 2	·529	·524	·519 5	·515	·513	·509	·504 5	·501
			Skin H.P.	...	687	...	999	...	1 396	1 621	1 881	2 153	2 465	2 800	3 220	3 530	3 958
			Resid. H.P.	...	164	...	279	...	516	577	699	960	1 516	2 500	3 845	5 380	6 742
			Resid. resistance lbs. per ton Δ	...	·584	...	·869	...	1·43	1·516	1·739	2·275	3·426	5·41	7·99	10·69	12·9

14B	29 200	6·6	⊙	...	·704	·705	·71	·724	·754	·767	·749	·75	·79	·871	·986	1·116	1·174
			$OSL^{-\cdot175}$	...	·56	·553	·546	·541	·534 8	·529 5	·525	·520 5	·516	·514	·51	·505 5	·501 5
			Skin H.P.	...	686	836	1 000	1·185	1 391	1 621	1 881	2 159	2 462	2 802	3 154	3 525	3 950
			Resid. H.P.	...	177	231	300	400	572	729	801	949	1 308	1 952	2 946	4 265	5 300
			Resid. resistance lbs. per ton Δ	...	·63	·768	·936	1·174	1·587	1·915	1·998	2·251	2·96	4·23	6·13	8·51	10·16
16B	30 300	6·84	⊙	...	·70	·70	·70	·71	·715	·742	·796	·816	·808	·855	·954	1·066	1·154
			$OSL^{-\cdot175}$	...	·56	·552 5	·546	·541	·534 8	·529 5	·525	·520 5	·516	·514	·51	·505 5	·501 5
			Skin H.P.	...	685	834	999	1 082	1 390	1 621	1 880	2 152	2 460	2 799	3 520	3 523	3 952
			Resid. H.P.	...	171	225	282	470	470	651	971	1 223	1 390	1 856	2 370	3 917	5 148
			Resid. resistance lbs. per ton Δ	...	·611	·75	·881	1·384	1·308	1·717	2·43	2·916	3·158	4·035	4·95	7·83	9·69
16D	30 290	6·845	⊙	...	·697	...	·706	...	·715	·74	·795	·815	·805	·857	·953	1·064	1·158
			$OSL^{-\cdot175}$	...	·56	...	·546 5	...	·535	·53	·525	·521	·516 5	·514	·51	·506	·502
			Skin H.P.	...	685	...	1 000	...	1 391	1 621	1 880	2 158	2 462	2 800	3 519	3 523	3 953
			Resid. H.P.	...	168	...	291	...	469	643	968	1 215	1 378	1 870	2 367	3 887	5 157
			Resid. resistance lbs. per ton Δ	...	·601	...	·911	...	1·306	1·698	2·422	2·9	3·134	4·07	4·94	7·78	9·92

Mr G. S. Baker's 1913 models, Set D. Model 28c. $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}25$. $\frac{\text{Length}}{\text{Beam}} = 7{\cdot}6$. Mid-area coef. = ·980. Prismatic coef.: entrance = ·672; run = ·638. Beam, 13·16 per cent. of length. Parallel body, 30 per cent. of length. Block coef. = ·739 5. Mean prismatic coef. = ·755. Ratio, $\frac{\text{Length entrance}}{\text{Length run}} = 1{\cdot}26$. $\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 161{\cdot}6$. $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \frac{\rho \times 427{\cdot}1}{\text{©}}$ where ρ is the propulsive coefficient.

Ⓚ	$\frac{V}{\sqrt{L}}$	ρ = probable propulsive coef. from naked model.	100 × 13·16 × 5·81. Δ = 161·6 tons.			200 × 26·32 × 11·61. Δ = 1 291 tons.			300 × 39·45 × 17·41. Δ = 4 360 tons.			400 × 52·6 × 23·21. Δ = 10 329 tons.			500 × 65·8 × 29. Δ = 20 200 tons.		
			Knots	©	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Knots	©	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Knots	©	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Knots	©	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	Knots	©	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
1·2	·480 5	·46	4·805	·786	250	6·8	·741	265	8·31	·714	275	9·61	·696	282	10·74	·688	288
1·3	·520 5	·466	5·205	·798	249·5	7·36	·753	264	9·01	·726	274	10·41	·708	281·5	11·63	·695	286
1·4	·560 5	·471 5	5·605	·812	248	7·91	·767	262·5	9·7	·740	272	11·21	·722	279	12·53	·709	284
1·5	·600 5	·476	6·005	·832	244·5	8·49	·787	258	10·4	·760	268	12·01	·742	274	13·41	·729	279
1·6	·640 5	·477	6·405	·844	242	9·05	·799	255	11·1	·772	264	12·81	·754	270	14·31	·741	275
1·7	·680 5	·478 5	6·805	·853	239·5	9·61	·808	253	11·8	·781	261	13·61	·763	267·5	15·21	·750	272
1·8	·720 5	·477 3	7·205	·885	230	10·19	·840	242·5	12·49	·813	251	14·41	·795	256·5	16·11	·782	261
1·9	·760 5	·472 5	7·605	·949	213	10·75	·904	223	13·19	·877	230	15·21	·859	235	17·0	·846	238·5
2·0	·800 5	·47	8·005	1·164	172	11·31	1·119	179·3	13·87	1·092	183·4	16·01	1·074	187	17·9	1·061	189

Mr G. S. Baker's models, 1913, Set A.

$L = \dfrac{V}{\sqrt{L}} \times 1{\cdot}055\,2$. $\quad S = \dfrac{S}{\Delta^{\frac{1}{3}}} \times {\cdot}093\,46$. $\quad$ Residuary resistance in lbs.

$= \dfrac{\text{Residuary H.P.}}{V \times {\cdot}003\,070\,7}$.

Model No.	Estimated wetted surface sq. ft.	Froude's S.	Results at various points.							
			$\frac{V}{\sqrt{L}}$	·379	·454 5	·568	·643 5	·757 5	·833	·870 5
			L	·400	·48	·60	·67	·80	·88	·919
14c	28 600	6·99	Ⓒ	·726	·72	·691	·709 95	·713 9	·81	·904
			$OSL^{-\cdot175}$	·608 5	·588 6	·566	·555 5	·539	·53	·525 5
			Skin H.P.	237	395	743	1 061	1 680	2 192	2 481
			Resid. H.P.	46	89	164	293	545	1 160	1 789
			Residuary resistance per ton Δ	·264 8	·426 1	·63	·992	1·569	3·031	4 47
14A	28 400	6·981	Ⓒ	·70	·681	·67	·664	·712	·741 2	·806
			$OSL^{-\cdot175}$	·609	·589	·567	·556	·538	·53	·525 5
			Skin H.P.	236	394	741	1 055	1 661	2 183	2 472
			Resid. H.P.	35·5	61·5	134	205	539	872	1 318
			Residuary resistance per ton Δ	·205 9	·297 3	·519	·701	1·561	2·304	3·32
29A	28 300	7·03	Ⓒ	·711 8	·686	·666	·674	·717 6	·729 5	·789 3
			$OSL^{-\cdot175}$	·608 5	·588	·566	·555	·537 5	·53	·526
			Skin H.P.	233	390·5	735	1 049	1 650	2 170	2 455
			Resid. H.P.	39·7	65·5	130	222	555	819	1 230
			Residuary resistance per ton Δ	·231 4	·318 1	·506	·761	1·617	2·17	3·12
16c	28 300	7·00	Ⓒ	·756	·743 5	·731 7	·726 3	·793	·818	·874
			$OSL^{-\cdot175}$	·61	·59	·568 6	·558	·54	·531 6	·528
			Skin H.P.	233	392	737	1 051	1 661	2 179	2 460
			Resid. H.P.	56	102	213	319	779	1 171	1 620
			Residuary resistance per ton Δ	·327 9	·499	·833	1·102	2·28	3·12	4·13

Mr G. Baker's models, Set A. Corrected for ships of constant length = 400 ft. b.p. Beam, 50 ft. Mean draught, 22·222 ft. Midship area coef. = ·980. $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}25$. $\frac{\text{Length}}{\text{Beam}} = 8$. Beam, 12·5 per cent. of length. Parallel body, 10 per cent of length. Prismatic coef.: entrance = ·52; run = ·584. $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ assumed = ·50 in calculating $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.

Model No.	Tons displacement.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.	Coefficients. Block.	Coefficients. Prismatic.	Estimated wetted surface. sq. ft.	Ratio. $\frac{\text{Length entrance}}{\text{Length run}}$	Results at various speeds. $\frac{V}{\sqrt{L}}$. / Knots.	·379 / 7·58	·454 5 / 9·09	·568 / 11·36	·643 5 / 12·87	·757 5 / 15·15	·833 / 16·66	·870 5 / 17·41
14C	7 480	116·9	·589	·601	28 600	·756	E.H.P.	283	484	907	1 354	2 225	3 352	4 270
							$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	294	297	309	301	299	264	236
14A	7 415	116	·584	·596	28 400	1·0 *	E.H.P.	271·5	455·5	875	1 260	2 200	3 055	3 790
							$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	305	313	318	322	300	287	265*
29A	7 374	115·3	·581	·593	28 300	1·323 *	E.H.P.	272·7	456	865	1 271	2 205	2 989	3 685
							$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	303	312	321	318	298	293	271*
16C	7 345	114·9	·579	·591	28 300	1·681	E.H.P.	289	494	950	1 370	2 440	3 350	4 080
							$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	285	288	292	294	270	261	245

$\text{E.H.P.} = \frac{\Delta^{\frac{2}{3}}}{427{\cdot}1} \times C \times V^3$.

* The most suitable for top speeds on trial.

On account of the slight differences in the displacement of the various models, the speed-length ratios and knots speed have not identical (K) values throughout. It is therefore necessary to interpolate the (C) values by drawing a curve. Thus for 16C, $V = \frac{\Delta^{\frac{1}{6}}}{{\cdot}583\,4} \times K = \frac{4{\cdot}408}{{\cdot}583\,4} \times K$; while for 14C, $V = \frac{4{\cdot}421}{{\cdot}583\,4} \times K$ For example, for (K) = 1 the speed with 16C is 7·56 knots, while with 14C it is 7·59 knots.

Mr G. S. Baker's models, 1913, Set C. Corrected for ships of 400 ft. b.p. Beam = 52·6 ft. Draught = 23·21. $\frac{\text{Beam}}{\text{Draught}} = 2·25$. Mid-area coef. = ·980. Prismatic coef.: entrance = ·57; run = ·584. $\frac{\text{Length}}{\text{Beam}} = 7·6$. Beam, 13·16 per cent. of length. Parallel body, 30 per cent. of length. $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ assumed = ·50 in calculating $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.

Model No.	Tons displacement.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Coefficients. Block.	Coefficients. Prismatic.	Ratio. $\frac{\text{Length entrance}}{\text{Length run}}$	Results at various speeds. $\frac{V}{\sqrt{L}}$	·435	·475	·553	·593	·633	·672	·711	·751	·790 5	·83	·87	·91
						Knots	8·7	9·5	11·06	11·86	12·66	13·44	14·22	15·02	15·81	16·6	17·4	18·2
18D	958 6	149·9	·686	·701	·554	E.H.P.	496	651	1 089	1 373	1 671	2 010	2 488	3 200	4 314	5 800	7 800	
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	299	297	281	274	274	272	261	239	206	178	152·5	
18C	958 1	149·8	·686	·700 5	·667	E.H.P.	500	658	1 053	1 360	1 663	1 965	2 362	2 900	3 689	4 760	6 650	
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	297	294	290	276	275	278	275	**263**	241	216	179	
18A	957 0	149·7	·685 5	·70	·905 *	E.H.P.	491	649	1 115	1 289	1 632	1 940	2 222	2 672	3 192	3 838	5 340	8 800
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	302	298	274	292	280	282	292	286	**279***	268	223	154·5
20A	956 4	149·6	·685	·699 6	1·212 *	E.H.P.	494	641	1 030	1 278	1 680	1 985	2 285	2 800	3 400	3 920	5 150	7 630
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	300	301	296	294	280	276	283	272	**262***	263	231	178
22A	955 2	149·4	·684 5	·699	1·627	E.H.P.	511	660	1 080	1 340	1 685	2 114	2 480	3 039	3 983	4 744	5 830	8 140
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	290	292	283	280	271	258	261	251	223	217	204	167

* Suitable top speeds on trial on measured mile in heavy type.

Mr G. S. Baker's models, 1913, Set C.—*continued.*

$L = \frac{V}{\sqrt{L}} \times 1{\cdot}055\,2$. $S = \frac{S}{\Delta^{\frac{2}{3}}} \times {\cdot}093\,46$, where S in the numerator is Taylor's wetted surface of the model without appendages. Residuary resistance in lbs. $= \frac{\text{Residuary H.P.}}{V \times {\cdot}003\,070\,7}$.

Model No.	Estimated wetted surface. sq. ft.	Froude's S.	Results at various speeds.												
			$\frac{V}{\sqrt{L}}$	·435	·475	·553	·598	·633	·672	·711	·751	·790 5	·83	·87	·91
			L	·459	·501	·584	·625 5	·669	·71	·750 5	·793	·834 5	·876	·919	·960 6
18D	32 280	6·69	©	·713 6	·719	·76	·78	·78	·785	·819 5	·895	1·034	1·2	1·4	
			$OSL^{-{\cdot}175}$	·568 5	·559	·545 3	·538	·532	·526	·521	·516	·511	·508	·503	·499 2
			Skin H.P.	395·5	506	780	946	1 140	1 348	1 581	1 846	2 133	2 445	2 800	
			Resid. H.P.	100·5	145	309	427	531	662	907	1 354	2 181	3 355	5 000	
			Resid. resistance lbs. per ton Δ	·392 2	·519	·949	1·225	1·429	1·675	2·169	3·061	4·69	6·87	9·76	
18C	32 270	6·69	©	·72	·726	·736	·772	·776	·766	·778	·81	·886	·986 5	1·194	
			$OSL^{-{\cdot}175}$	·568 5	·559	·545 3	·538	·532	·526	·521	·516	·511	·508	·503	
			Skin H.P.	394·2	506	781	947	1 140	1 349	1 588	1 849	2 129	2 451	2 799	
			Resid. H.P	105·8	152	272	413	523	616	779	1 051	1 560	2 309	3 851	
			Resid. resistance lbs. per ton Δ	·412 5	·544	·836	1·185	1·403	1·559	1·861	2·381	3·355	4·726	7·53	

18A	32 260	6·69	ⓒ	·706	·718	·71	·732	·763	·758	·732	·747	·766	·795	·96	1·384
			$OSL^{-·175}$	·568 5	·559	·545 3	·538	·532	·526	·521	·516	·511	·508	·503	·499 2
			Skin H.P.	394·6	505	856	946	1 139	1 347	1 582	1 846	2 129	2 450	2 795	3 175
			Resid. H.P.	96·4	144	259	343	493	593	640	826	1 063	1 388	2 545	5 625
			Resid. resistance lbs. per ton Δ	·376 9	·515 5	·797	·985	1·327	1·50	1·53	1·872	2·286*	2·845	4·98	10·52
20A	32 250	6·692	ⓒ	·711	·708	·72	·726	·762	·775	·754	784	·815	·813	·926	1·2
			$OSL^{-·175}$	·568 7	·560	·546	·538 6	·532 5	·526 3	521	·516	·513	·508 4	·504	·500
			Skin H.P.	394·2	508	781	946	1 139	1 349	1·580	1 843	2 139	2 450	2 800	3 180
			Resid. H.P.	100	133	249	332	491	636	705	957	1 261	1 470	2 350	4 450
			Resid. resistance lbs. per ton Δ	·391 5	·477 4	·767	·954	1·321	1·61	1·687	2·168	2·716*	3·014	4·6	8·32
22A	32 250	6·695	ⓒ	·738	·73	·756	·763	·788	·826	·818	·854	·958	·984	1·049	1·28
			$OSL^{-·175}$	·569	·560	·546	·540	·533	·527	·521 6	·516	·513	·509	·504 2	·500
			Skin H.P.	394	506	780	950	1 140	1 349	1 581	1 837	2 138	2 455	2 800	3 180
			Resid. H.P.	117	154	300	390	545	765	899	1 202	1 845	2 289	3 030	4 960
			Resid. resistance lbs. per ton Δ	·459	·552	·925	1·121	1·469	1·939	2·152	2·728	3·973	4·70	5·93	9·30

* Suitable on trial.

Mr G. S. Baker's models, 1913, Set D.—*continued.*

(a) values obtained by scaling the ordinates of Mr Baker's curves. $L = \frac{V}{\sqrt{L}} \times 1{\cdot}055\,2$. $S = \frac{S}{\Delta^{\frac{2}{3}}} \times {\cdot}093\,46$, where S in the numerator is Taylor's wetted surface of the model without appendages. Residuary resistance in lbs. $= \frac{\text{Residuary H.P.}}{V \times {\cdot}003\,070\,7}$.

Model No.	Estimated wetted surface in sq. ft.	Froude's S.	Results at various speeds.											
			$\frac{V}{\sqrt{L}}$	·440 5	·480 5	·520 5	·560 5	·600 5	·640 5	·680 5	·720 5	·760 5	·800 5	·841
			L.	·465	·507	·550	·591	·633 5	·676	·719	·76	·803	·845	·888
23A	33 460	6·62	(c)	·672	·682	·700	·706	·734	·776	·87	1·012	1·234	1·457	
			$OSL^{-\cdot175}$	561	·553	·546	·539	·532	·526	·520	·514 5	·510	·505	
			Skin H.P.	433	542	681	839	1 019	1 222	1 450	1 701	1 984	2 290	
			Resid. H.P.	85	127	192	260	386	582	976	1 648	2 812	4 315	
			Resid. resistance, lbs. per ton Δ	·314	·43	·600	·755	1·048	1·48	2·337	3·72	6·02	8·78	
23B	33 500	6·618	(c)	·67	·679	·693	·714	·716	·736	·758	·854	1·058	1·256	1·344
			$OSL^{-\cdot175}$	·561	·553	·546	·539	·532	·526	·520	·514 2	·510	·505	·501
			Skin H.P.	425	·544	·683	839	1 020	1 224	1 450	1 703	1 983	2 298	2 640
			Resid. H.P.	82	123	183	272	353	488	666	1 127	2 129	3 412	4 440
			Resid. resistance, lbs. per ton Δ.	·308	·416	·572	·790	·957	1·24	1·593	2·548	4·55	6·94	8·6

19A	33 510	6·618	(c)	·687	·691	·696	·725	·78	·741 5	·804	·973	1·112	1·142	
			$OSL^{-·175}$	·561	·553	·546	·539	·532	·526	·520	·514 2	·510	·505	
			Skin H.P.	426	544	680	840	1 020	1 225	1 450	1 704	1 984	2 295	
			Resid. H.P.	95	136	187	291	380	502	792	1 521	2 346	2 895	
			Resid. resistance, lbs. per ton Δ	·351	·460	·584	·845	1·002	1·278	1·896	3·435	5·025	5·89	
23C	33 550	6·61	(c)	·692	·696	·708	·722	·742	·754	·763	·795	·859	1·074	1·078
			$OSL^{-·175}$	·560 8	·552	·545	·538 2	·531 5	·526	·519 5	·514	·509 8	·505	·500 5
			Skin H.P.	·426	·544	681	839	1 021	1 227	1 456	1 707	1 987	2 296	2 640
			Resid. H.P.	99	142	203	289	405	532	682	933	1 361	2 589	3 043
			Resid. resistance, lbs. per ton Δ	·366	·481	·635	·840	1·098	1·352	1·631	2·108	2·915	5·26	5·89
23D	33 580	6·61	(c)	·734	·753	·745	·766	·800	·812	·858	·914	1·052	1·21	1·152
			$OSL^{-·175}$	·560 8	·552	·545	·538·2	·531 5	·526	·519 5	·514	·509 8	·505	·500 5
			Skin H.P.	426	545	684	840	1 024	1 230	1 457	1 707	1 991	2 300	2 642
			Resid. H.P.	131	198	250	358	516	670	947	1 828	2 123	3 210	3 448
			Resid. resistance, lbs. per ton Δ.	·483	·671	·781	1·04	1·399	1·703	2·268	3·00	4·55	6·52	6·67

Mr G. S. Baker's models, 1913, Set D. Corrected for ships of 400 ft. b.p. Beam = 52·6 ft. Draught = 23·21 ft. $\frac{\text{Beam}}{\text{Draught}}$ = 2·25. Mid-area coef. = ·980. Prismatic coef.: entrance = ·672; run = ·638. $\frac{\text{Length}}{\text{Beam}}$ = 7·6. Beam 13·16 per cent of length. Parallel body, 30 per cent. of length. $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ assumed = ·50 in calculating $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$.

Model No.	Tons displacement.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Coefficients. Block.	Coefficients. Prismatic.	Ratio. $\frac{\text{Length entrance}}{\text{Length run}}$	$\frac{V}{\sqrt{L}}$ / Knots	·440 5 / 8·81	·480 5 / 9·61	·520 5 / 10·41	·560 5 / 11·21	·600 5 / 12·01	·640 5 / 12·81	·680 5 / 13·61	·720 5 / 14·41	·760 5 / 15·21	·800 5 / 16·01	·841 / 16·82
23A	10 268	160·4	·735	·75	·599	E.H.P.	518	669	873	1 099	1 405	1 804	2 426	3 349	4 796	6 605	
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	312	313	305	303	291	276	246	211	173	147	
23B	10 293	160·9	·736	·751	·800	E.H.P.	507	667	866	1 111	1 373	1 712	2 116	2 880	4 112	5 710	7 080
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	319	315	308	300	298	290	282	250	202	170	159
19A	10 310	161·1	·739	·754	1·00	E.H.P.	521	680	867	1 131	1 400	1 727	2 242	3 225	4 330	5 190	
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	310	308	307	295	293	288	266	219	192	187	
23C	10 329	161·6	·739 5	·755	1·26	E.H.P.	525	686	884	1 128	1 426	1 759	2 138	2 640	3 348	4 885	5 683
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	308	306	302	296	287	283	279	268	249	199	198
23D	10 346	161·8	·740	·755	1·668	E.H.P.	557	743	934	1 198	1 540	1 900	2 404	3 035	4 114	5 510	6 090
						$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$	291	316	287	279	267	263	249	234	203	177	186

$V = \frac{\Delta^{\frac{1}{6}}}{\cdot 5834} \times K$. E.H.P. $= \frac{\Delta^{\frac{2}{3}}}{27 \cdot 1} \times C \times V^3$. Suitable top speed on measured mile on trial in heavy type.

Mr G. S. Baker's models, Set E. Corrected for ships of constant length = 400 ft. b.p. Beam = 52·6. Draught = 23·21. $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}25$. Mid-area coef. = ·98. Prismatic coef.: entrance = ·672; run = ·638. $\frac{\text{Length}}{\text{Beam}} = 7{\cdot}6$. Beam, 13·16 per cent. of length. Parallel body, 50 per cent. of length. $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ assumed = ·50 in calculating $\frac{\Delta^{2/3}V^3}{\text{I.H.P.}}$.

Model No.	Tons displacement.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Coefficients. Block.	Coefficients. Prismatic.	Ratio: $\frac{\text{Length entrance}}{\text{Length run}}$	$\frac{V}{\sqrt{L}}$ / Knots	·405 5 / 8·11	·446 / 8·92	·486 5 / 9·73	·527 / 10·54	·568 / 11·36	·608 5 / 12·17	·649 5 / 12·99	·690 / 13·8	·730 / 14·6	·770 5 / 15·41
19E	11 220	175·5	·808	·810 4	·602	E.H.P.	467	626	861	1 138	1 536	2 090	2 910	4 310	5 400	7 400
						$\frac{\Delta^{2/3}V^3}{\text{I.H.P.}}$	286	284	268	258	239	216·5	189	153	144·6	
19D	11 232	175·6	·804	·82	·738	E.H.P.	488	643	848	1 116	1 466	1 973	2 780	3 880	4 960	7 000
						$\frac{\Delta^{2/3}V^3}{\text{I.H.P.}}$	274	277	272	263	251	**229**	198	170	157·5	131
19B	11 259	175·8	·805	·821 4	·999	E.H.P.	489	649	855	1 104	1 448	1 847	2 658	3 570	4 390	6 740
						$\frac{\Delta^{2/3}V^3}{\text{I.H.P.}}$	274	274	270	266	254	**245**	207	185	178	136
24B	11 266	175·9	·805 5	·821 9	1·247	E.H.P.	514	680	895	1 164	1 520	1 940	2 660	3 585	4 332	6 850
						$\frac{\Delta^{2/3}V^3}{\text{I.H.P.}}$	261	262	258	252	242	233	207	184	180·4	134
24A	11 276	176	·806	·822 4	1·653	E.H.P.	582	803	1 062	1 379	1 850	2 388	3 276	4 086	4 812	8 781
						$\frac{\Delta^{2/3}V^3}{\text{I.H.P.}}$	230	222	218	214	199	190	**168**	162	162·5	104·7

Suitable top speed on measured mile on trial in heavy type.

Mr G. S. Baker's Models, 1913. Set E.—*continued.*

$$L = \frac{V}{\sqrt{L}} \times 1{\cdot}055\,2. \quad S = \frac{S}{\Delta^{\frac{2}{3}}} \times {\cdot}093\,46. \quad \text{Residuary resistance in lbs.} = \frac{\text{Residuary H.P.}}{V \times {\cdot}003\,070\,7}.$$

Model No.	Estimated wetted surface in sq. ft.	Froude's S.	Results at various speeds.										
			$\frac{V}{\sqrt{L}}$.	·405 5	·446	·486 5	·527	·568	·608 5	·649 5	·690	·730	·770 5
			L.	·428 2	·471	·514	·556	·600	·642	·685 5	·729	·771	·814
19E	34 950	6·52	(C)	·746	·752	·796	·828	·892	·988	1·132	1·396	1·48	1·736
			$OSL^{-{\cdot}175}$	·560 5	·551	·543	·536	·529	·522	·516	·510 5	·505	·500
			Skin H.P.	351	459	586·6	736	910	1 105	1 328	1 574	1 842	2 146
			Resid. H.P.	116	167	274	402	626	985	1 582	2 736	3 558	5 294
			Resid. resistance, lbs. per ton Δ.	·415	·543	·942	1·106	1·599	2·348	3·54	5·746	7·07	9·95
19D	35 000	6·515	(C)	·78	·771	·784	·812	·853	·933 6	1·08	1·258	1·358	1·631
			$OSL^{-{\cdot}175}$	·56	·550 5	·543	·536	·528 5	·522	·516	·510	·505	·500
			Skin H.P.	350	459	586·8	736	908	1 103	1 409	1 575	1 845	2 146
			Resid. H.P.	138	184	261	380	558	870	1 371	2 305	3 115	4 854
			Resid. resistance, lbs., per ton Δ.	·493	·598 2	·777 5	1·046	1·423	2·072	3·062	4·84	6·18	9·12

19B	35 020	6·52	C	·76	·778	·79	·802	·84	·873	1·03	1·156	1·2	1·568
			$OSL^{-\cdot 175}$	·560 5	·551	·543	·536	·529	·522	·516	·510 5	·505	·500
			Skin H.P.	360·8	460	587	738	911	1 105	1 331	1 578	1 850	2 150
			Resid. H.P.	128	189	268	366	537	742	1 327	1 992	2 540	4 590
			Resid. resistance, lbs. per ton Δ.	·456 4	·613 5	·798	1·006	1·371	1·764	2·959	4·183	5·04	8·61
24B	35 050	6·52	C	·82	·816	·826	·846	·882	·916	1·033	1·162	1·184	1·592
			$OSL^{-\cdot 175}$	·560 5	·551	·543	·536	·529	·522	·516	·510 5	·505	·500
			Skin H.P.	351·5	459	588·5	788	911	1 107	1 329	1 575	1 847	2 151
			Resid. H.P.	162·5	221	306·5	426	609	838	1 331	2 010	2 485	4 699
			Resid. resistance, lbs. per ton Δ.	·579	·716	·913	1·17	1·553	1·981	2·969	4·21	4·93	8·81
24A	35 080	6·518	C	·928	·962	·98	1·00	1·072	1·128	1·27	1·32	1·316	2·038
			$OSL^{-\cdot 175}$	·560 5	·551	·543	·536	·529	·522	·516	·510 5	·505	·500
			Skin H.P.	352	460	588·5	739	911	1 108	1 331	1 580	1 850	2 155
			Resid. H.P.	230	343	473·5	640	939	1 280	1 945	2 506	2 962	6 626
			Resid. resistance, lbs. per ton Δ.	·869	1·111	1·406	1·756	2392	3 041	4·34	5·25	5 865	12·42

Sadler, *Transactions American Society Naval Architects and Marine Engineers*, 1915. Ships about 400 ft. in length derived from the particulars given in the above paper. The resistances are scaled from the curves published in *Engineering*.

Type No. 1 (B).	$\frac{L}{B}$.	Beam as percentage of length.	$\frac{L}{D}$.	$\frac{B}{D}$.	Dimensions.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Coefficients. Mid area.	Prismatic.	Block.	Δ tons.	Residuary resistance in lbs. per ton Δ at various speeds $\frac{V}{\sqrt{L}}$. ·50	·60	·70	·80	·85	·90	·95	1·00	1·05	1·10
Figs. 2 and 3	9·02	11·1	18	2·0	400×44·4̇×22·2̇	108	·92	·665	·612	6 910	·70	1·15	1·75	2·4	3·05	4·1	6·5	10·0		
Figs. 2 and 3	7·2	13·9	18	2·5	400×55·5̇×22·2̇	135	·92	·665	·612	8 650	·80	1·4	1·9	2·8	3·6	4·7	7·5	11·5		
Figs. 2 and 3	6·0	16·6̇	18	3·0	400×66·6̇×22·2̇	162	·92	·665	·612	10 380	·90	1·5	2·2	3·2	4·05	5·35	8·4	12·9		
Figs. 2 and 3	7·2	13·9	20	2·7̇	400×55·5̇×20	121·8	·92	·665	·612	7 770	·90	1·4	2·0	2·96	3·8	5·0	7·85	12·1		
Figs. 2 and 3	7·2	13·9	22·5	3·125	400×55·5̇×17·8	108	·92	·665	·612	6 910	·92	1·4	2·04	3·0	3·9	5·1	8·05	12·5		
Fig. 4	6·48	15·45	16·2	2·5	360×55·5̇×22·2̇	166·7	·92	·665	·612	7 770	·75	1·2	1·9	2·86	3·55	4·7	7·45	11·7		
Fig. 4	7·92	12·63	18·8	2·5	440×55·5̇×22·2	111·9	·92	·665	·612	9 505	·65	1·12	1·83	2·55	3·2	4·4	6·96	10·75		
Fig. 5	7·2	13·9	18	2·5	360×50	..	..	·649	·596	6 130	1·0	1·5	2·45	3·5	..	7·5	..	15·7	..	21·5
Type 2 (B)																				
Figs. 2 and 3	10·0	10·0	20	2·0	400×40×20	76·7	·92	·598	·537	4 910	·60	·90	1·16	1·7	2·2	3·1	4·8	6·6	7·92	9·15
Figs. 2 and 3	8·0	12·5	20	2·5	400×50×20	95·8	·92	·598	·537	6 130	·70	1·0	1·3	2·0	2·6	3·6	5·5	7·7	9·0	10·45
Figs. 2 and 3	6·6̇	15·0	20	3·0	400×60×20	115·1	·92	·598	·537	7 370	·90	1·1	1·66	2·26	2·92	4·0	6·2	8·6	10·1	11·8

Ships about 400 ft. in length, derived from particulars given in Prof. Sadler's paper in the *Transactions American Society of Naval Architects and Marine Engineers*, 1915. The resistances are scaled from the curves published in *Engineering*.

Type No. 2 (B).	$\frac{L}{B}$.	Beam as percentage of length.	$\frac{L}{D}$.	$\frac{B}{D}$.	Dimensions.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Coefficients.			Displacement in tons.	Residuary resistance in lbs. per ton Δ at various speeds $\frac{V}{\sqrt{L}}$.									
							Mid area.	Pris-matic.	Block.		·50	·60	·70	·80	·85	·90	·95	1·00	1·05	1·10
Figs. 2 and 3	8·0	12·5	$22{\cdot}\dot{2}$	$2{\cdot}\dot{7}$	400×50×18	86·4	·92	·598	·537	5 535	·70	1·0	1·35	2·0	2·65	3·8	5·8	8·05	9·4	11·0
Figs. 2 and 3	8·0	12·5	25	3·125	400×50×16	76·7	·92	·598	·537	4 910	·75	1·04	1·5	2·1	2·75	3·9	5·92	8·35	9·75	11·25
Fig. 4	7·2	13·9	18	2·5	360×50×20	118·4	·92	·598	·537	5 535	·75	1·0	1·5	2·1	2·82	3·8	5·9	8·0	9·4	11·0
Fig. 4	8·8	11·38	22	2·5	400×50·20	79·4	·92	·598	·537	6 760	·75	1·0	1·35	1·9	2·4	3·5	5·3	7·5	8·65	10·0
3 (B) Figs. 2 and 3	11·0	..	22	2·0	$400\times36{\cdot}\dot{3}\dot{6}\times18{\cdot}\dot{1}\dot{8}$	..	·92	·544	·501	..	·75	1·0	1·3	1·7	2·0	2·66	3·02	3·75	4·45	5·6
Figs. 2 and 3	8·8	..	22	2·5	$400\times45{\cdot}\dot{4}\dot{5}\times18{\cdot}\dot{1}\dot{8}$	..	·92	·544	·501	..	·85	1·1	1·5	2·0	2·3	3·0	3·5	4·25	5·08	6·45
Figs. 2 and 3	$8{\cdot}\dot{3}$	..	22	2·64	$400\times48\times18{\cdot}\dot{1}\dot{8}$	..	·92	·544	·501	..	·95	1·25	1·7	2·15	2·6	3·4	4·0	4·9	5·75	7·1
Figs. 2 and 3	8·8	..	$24{\cdot}\dot{4}$	$2{\cdot}\dot{7}$	$400\times45{\cdot}\dot{4}\dot{5}\times16{\cdot}\dot{3}\dot{6}$	..	·92	·544	·501	..	·80	1·0	1·4	2·0	2·35	3·0	3·75	4·5	1·27	6·6
Figs. 2 and 3	8·8	..	27·5	3·125	$400\times45{\cdot}\dot{4}\dot{5}\times14{\cdot}\dot{5}\dot{4}$	..	·92	·544	·501	..	·80	1·1	1·5	2·0	2·4	3·13	3·9	4·7	5·4	6·85
Fig. 4	7·92	..	19·8	2·5	$360\times45{\cdot}\dot{4}\dot{5}\times18{\cdot}\dot{1}\dot{8}$	..	·92	·544	·501	..	·95	1·3	1·5	2·0	2·5	3·2	3·85	4·7	5·6	6·98
Fig. 4	9·68	..	$24{\cdot}\dot{2}\dot{4}$	2·5	$440\times45{\cdot}\dot{4}\dot{5}\times18{\cdot}\dot{1}\dot{8}$	..	·92	·544	·501	..	·80	1·1	1·4	1·9	2·27	2·9	3·6	4·3	5·0	6·41
Fig. 5	8·8	11·38	22	2·5	440×50×20	..	..	..	·539	6 760	·80	1·0	1·1	1·7	..	2·5	..	3·6	..	5·0

Ships about 400 ft. in length, derived from particulars given in Prof. Sadler's paper to the *American Society of Naval Architects and Marine Engineers*, 1915. The resistances are scaled from the curves published in *Engineering*.

Type No.	$\frac{L}{B}$.	Beam as percentage of length.	$\frac{L}{D}$.	$\frac{B}{D}$.	Dimensions.	$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$	Coefficients. Mid area.	Coefficients. Prismatic.	Coefficients. Block.	Tons Δ.	Residuary resistance in lbs. per ton Δ at various speeds $\frac{V}{\sqrt{L}}$. ·50	·60	·70	·80	·85	·90	·95	1·00	1·05	1·10
Fig. 5 1 (A)	9·0	11·11	18	2·0	360×40×20	111	·936	·674	·63	5 190	·91	1·4	2·25	3·15	..	6·5	..	18·8	..	19·0
2 (A)	10·0	10·0	20	2·0	400×40×20	81	·936	·606	·567	5 190	·75	1·0	1·4	1·8	..	3·25	..	6·5	..	9·15
3 (A)	11·0	9·1	22	2·0	440×40×20	60·9	·936	·551	·515	5 190	·70	·90	1·1	1·6	..	2·25	..	3·45	..	5·0
1 (C)	6·0	16·6̇	18	3·0	360×60×20	155	·904	·648	·585	7 220	1·1	1·75	2·7	3·9	..	8·0	..	17·2	..	23·8
2 (C)	6·6̇	15·16	20	3·0	400×60×20	112·8	·904	·583	·526	7 220	·90	1·25	1·6	2·15	..	4·0	..	8·05	..	11·5
3 (C)	7·3̇	13·6̇	22	3·0	440×60×20	84·7	·904	·530	·479	7 220	·85	1·0	1·25	1·8	..	2·5	..	3·7	..	5·2
1 (B)	7·2	13·9	18	2·5	360×50×20	..	·92	..	·597	6 140										
2 (B)	8·0	12·5	20	2·5	400×50×20	..	·92	·598	·537	6 140										
3 (B)	8·8	11·38	22	2·5	440×50×20	..	·92	..	·488	6 140										

Independent Estimate of Power for Propulsion.

The I.H.P. or S.H.P. may be built up thus :—

(1) The E.H.P. of the naked hull is got from a tank trial or calculated from (*a*) the skin H.P., and (*b*) the residuary H.P. from Taylor's contours of residuary resistance per ton of displacement. It is often considered advisable to add 5 per cent. to Taylor's figures, because the temperatures at the U.S.A. tank are higher on the average, and show lower resistance, than those of general practice.

(2) A percentage is added for appendage resistance ; this may be taken from Captain Dyson's figures.

(3) The air H.P. is added.

Thus we have $\dfrac{\text{E.H.P. (naked)}+\text{appendages}+\text{air H.P.}}{\text{Hull efficiency}}=\text{T.H.P.}$

The hull efficiency (e_3) is theoretically the factor which provides for the effect of the proximity of the propeller to the hull.

(4) The D.H.P. = power delivered to the propeller $=\dfrac{\text{T.H.P.}}{\text{Screw efficiency}}$. Propeller efficiencies may be taken from Mr R. E. Froude's results, shown on our Plates 55–63.

(5) The S.H.P.

$$=\text{Shaft horse-power}=\frac{\text{D.H.P.}}{\text{Shaft transmission efficiency}}.$$

The shaft transmission efficiency, which may be taken from Plate 41, differs from the D.H.P. by the amount of friction in the stern tube and tunnel bearings.

(6) The I.H.P. is greater than the S.H.P. by the amount of friction in the engine itself when it is a reciprocating engine. The S.H.P. is the power taken at the aft end of the thrust shaft, while the D.H.P. is the power at the outer end of the stern tube.

Plate 41 shows ratios of D.H.P. to I.H.P. and S.H.P. from Messrs Maclaren and Welsh's paper (*Trans. Inst. Engineers and Shipbuilders, Scot.*, 1914).

$$\text{Propulsive coefficient}=\frac{\text{E.H.P. (naked)}}{\text{S.H.P. or I.H.P.}}.$$

The T.H.P. may be taken as

$$\frac{\text{E.H.P. (naked)}+\text{a percentage addition for appendages}}{\text{Hull efficiency}}$$

and air H.P. taken separately.

Analysis of trial-trip results and of propeller performances on actual service, in cases where model of the ship has not been tried. The E.H.P. is estimated, the skin H.P. being calculated and the residuary H.P. obtained from Taylor's contours and the air resistance calculated, and additional power for appendage resistance taken from Dyson's book, and engine friction and propeller waste from our Plates 37 and 40, based upon Maclaren and Welsh's 1914 curves.

Usually a wake value is assumed, using figures from Baker, Froude, Luke, MacDermott or Taylor. The propeller efficiency, which may be taken from Taylor's experiments or from T. B. Abell's 1910 paper, depends on real slip ratio, which is known if we assume a wake value. So that we take approximately

$$\frac{\text{E.H.P. (naked model)}}{\text{D.H.P.} \times e_2} = \text{hull efficiency} = e_3.$$

The hull efficiency includes all the unknown quantities, and can only be estimated from a similar ship.

From trial-trip results hull efficiencies on this basis vary from ·80 to 1·0. The lower figure applies to small twin-screw ships and the higher figure to large twin-screw passenger liners; the reason for the difference is at present obscure. In single screws 1·3 may be found. If Baker's allowance for effect of form upon frictional resistance be correct, the method of estimating power from Taylor's contours must be considerably affected, though in many cases, as for instance in the example on p. 77, Taylor's residuary resistance is low and agrees with this method. Taylor, however, did not make this allowance. On actual service, of course, the propeller efficiency will be low and the slip ratio high as compared with trial-trip results.

For *Air Resistance*, the formula KV^2.

If the resistance is expressed in tons, and V in tens of knots, then for the "Powerful" K = ·5, "Vulcan" = ·3, "Medusa" = ·15. Suppose that for a given vessel it had been calculated that there was about 4 000 sq. ft. of surface above the L.W.L., reckoned normal to the direction of motion.

$$\text{Pressure per sq. ft.} = \frac{v^2}{330} \text{ lbs. } (v \text{ in miles per hour}).$$

$$\text{At 20 knots pressure per sq. ft.} = \frac{\left(20 \times \frac{6\,080}{550}\right)^2}{330} = 1{\cdot}61 \text{ lbs.}$$

$$\text{Horse-power absorbed} = \frac{1{\cdot}61 \times 4\,000 \times 20}{33\,000} \times \frac{6\,080}{60} = 395.$$

Thrust Horse-power, T.H.P.—This is the basis figure for all propeller calculations. It may be arrived at either (1) from the resistance of the ship, or (2) from the propeller performance.

(1) To the E.H.P. (naked) an addition is made for wind resistance and for appendage resistance. The total E.H.P. thus found is divided by the hull efficiency, and the quotient is the T.H.P.

(2) From the wake value, the ship speed, and the revolutions the whole propeller performance, including the thrust horse-power, can be worked out.

The T.H.P. from (1) should equal the T.H.P. from (2) if all the values are correct, but they almost never agree; (1) is usually about 10 per cent. less than (2). In most cases this is because too little has been allowed for air resistance, and perhaps too little for appendage resistance.

Suppose in (1) we have propulsive efficiency stated as ·50, in (2) we have engine efficiency ·84, hull efficiency ·98, propeller efficiency ·70, air and appendage factor ·91, these giving a product of ·525, this is a difference of 5 per cent.

If the wind resistance is calculated from the areas by the formula, it will be found greater than is usually guessed, and the discrepancy will then be much reduced.

A set of curves of $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ or $\frac{\text{E.H.P.}}{\text{S.H.P.}}$ for different types of ships, taken from actual running, should be obtained and kept up to date. The E.H.P. (naked), from tank trial, which is given in comparatively few cases, may be replaced by E.H.P. calculated from Taylor's contours of residuary resistance per ton of displacement, and our tables of skin H.P. per 1 000 square feet of wetted surface. To the latter we should add a percentage, 5 per cent. or so, which we may call Mr Baker's addition for form. To Mr Taylor's residuary resistance about 5 per cent. should also be added to bring the relatively warm-water results of the American experimental results into line with average sea temperatures. I.H.P. and S.H.P. include appendage additions, which amount to about 4 per cent. for single screws and 9 per cent. for twin screws. Sea speeds may be taken as ·925 of trial speeds at the same power, for medium-sized vessels, the reduction being due principally to wind effects. Professor Durand mentions that wind resistance amounts to 25 per cent. of water resistance for 10 knots against a 40-knot wind.

For converting trial speeds and trial-trip values of $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$ into sea-going figures, a wind velocity of 20 knots may be taken for

the calculation. For little ships, battling against waves, the sea speed is lower compared with their trial speed, while with large vessels the trial speed and the sea speed are much alike, because the large vessels are less susceptible to the opposing forces of weather and sea.

"Wind Pressure on Ships" is the subject of an article in *Der Schiffbau*, an abstract of which was given in the *Shipbuilding and Shipping Record*, 24th April 1917. This article condemns the usual formula which only takes account of the transverse area of the exposed surface, and considers the increase of velocity due to height, comparing the influence of the fine lines of the "Mauretania" with that of the blunter lines of the passenger and cargo liner "Kaiserin Auguste Victoria," pointing out from deck to deck how everything in the former was planned with a view to lessening wind resistance. The pressure on the funnels, masts, and other curved portions of the vessel are calculated, and the average velocity of resistance to wind of the anchored ship, taking into the calculation rail supports, horizontal friction surfaces, cable-stoppers, windlass, capstans, davits, bollards, etc.

If we have E.H.P. curves from tests of tank models of a few ships, curves of values of (C) may be plotted, and from these a new curve of length-correction for (C) may be derived, similar to Mr Baker's, except that it will be steeper on account of sea and weather effect upon small ships, making (C) a more useful quantity.

T.S.S. "H.," $440 \times 54{\cdot}1 \times 23{\cdot}5$ ft. mean draught. Block coef. = ·637. 14½ knots at sea. 85 revs. 5 300 I.H.P. at sea.

From Taylor's curves, E.H.P. (naked) = 2 236.
From tank trial E.H.P. (naked) = 2 470.

$$\text{Taking 5 300 I.H.P. we have } \frac{\text{E.H.P.}}{\text{I.H.P.}} = \frac{2\,470}{5\,300} = \cdot 466.$$

$$\text{and } \frac{\text{E.H.P.}}{\text{I.H.P.}} = \frac{2\,236}{5\,300} = \cdot 422.$$

·422 is the "nominal efficiency of propulsion," at sea, and ·466 is the "propulsive coefficient" from tank-model results.

Taylor's contours are invaluable for providing the means for making a set of "nominal propulsive efficiencies" from the performances of known ships of various types, upon which an estimator may base calculations for the power of proposed ships. It does not matter though the calculated E.H.P. (naked) and the "nominal propulsive coefficient" be considerably lower than the

·50 usually accepted as a standard, so long as we keep to the same method of arriving at the result for the proposed vessel as for the type ships.

Messrs Maclaren and Welsh's vessel A. Single-screw steamer or yacht, with three-crank triple-expansion reciprocating steam engine. 14 knots on trial. 169 × 26 × 8·45 ft. trial draught. Δ = 573 tons. Block coef. = ·54. Prism. coef. = ·59. Mid-area coef. = ·915.

Knots.	I.H.P.	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}}$.	Percentage of fourteen knots.
10	282	245	71·5
11	394	268	78·6
12	516	265	85·8
13	696	250	93
14	962	225	100

Our curve of appropriate $\frac{V}{\sqrt{L}}$ for this form gives 12·7 knots. Therefore for speeds at sea we should plot the following, taking corresponding values of $\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}}$ for the same percentages of the service speed of 12·7 knots.

Knots.	Percentage of 12·7 knots.	I.H.P.	$\frac{\Delta^{\frac{2}{3}} V^3}{\text{I.H.P.}}$.	$\frac{\text{E.H.P.}}{\text{I.H.P.}}$ (naked model) or propulsive coefficient.
9·08	71·5	282	209	·44
9·99	78·6	394	200	·45
10·9	85·8	516	200	·457
11·8	93	696	187	·457
12·7	100	962	168	·45

$$\frac{12{\cdot}7}{14} = {\cdot}907.$$

Appendage Resistance.

Captain C. W. Dyson, of the U.S. Navy, considers that the resistances of the bilge keels, docking keels, shafting, struts and shaft bosses, are skin-frictional, and can be calculated as such, while the rudder, stern post, and scoops if any, enter more into eddy-making resistance. The percentage addition for the latter, therefore, is subject to Froude's Law of Comparison for want of a better method. In his book *Screw-Propellers and Estimation of Power for Propulsion of Ships*, Captain Dyson bases his diagram for appendage resistance percentage additions upon the assumption that these vary directly as beam-as-percentage-of-length of ship, taking a standard block coefficient of ·60.*

Model experiments are in almost all cases made with the bare or naked hull only, and this may be supposed to include a reasonable amount of deadwood. Any excess deadwood adds to the skin-frictional resistance.

Instead of adding the percentage increase for the resistance of the appendages taken altogether to the total E.H.P. as Captain Dyson does, we prefer to separate those which increase the skin-frictional resistance from the group of appendages which affect the eddy-making resistance, in the manner indicated on p. 5.

In a paper by Mr T. G. Owens, read before the Inst. Naval Architects in 1914, it was noted that the resistance results of rudder appendages deduced from experiments with models were somewhat exaggerated, and that twin rudders adversely affected the value of the propulsive coefficient to a considerable extent. In the discussion, Sir Philip Watts said that the increase in power required in passing from middle-line rudders to side rudders at the same speed was about 3 per cent. of the whole horse-power, with properly shaped appendages and rudders of only equal power, and that for that reason twin-side rudders had been given up in British Dreadnoughts and in certain foreign warships, in spite of the advantage, with side rudders, of being able to turn a vessel quickly even when stationary when the screws are driven hard ahead, because the loss of speed entailed was about a quarter of a knot on a 25-knot ship.

A four-propeller ship has more appendage resistance due to the shafts than a three-propeller ship.

* Corrective curves are given, showing decreasing appendage resistance for fuller ships, and slightly increasing percentages for finer forms.

APPENDAGES.

Single-screw vessels.	Twin-screw vessels. Triple-screw and four-shaft vessels.
Rudder. Rudder post. Bilge keels. Shaft bossing (negligible). Propeller boss (only if unusually large).	Rudder or rudders. Rudder post. Bilge keels. Docking keels (in very large vessels). Shafts. Struts. Shaft bossings. Deadwood (if over a reasonable amount). Scoops (if any are fitted).

Some examples showing percentage additions for the increase of resistance due to appendages, taken from Captain Dyson's book, *Screw-Propellers and Estimation of Power for Propulsion of Ships*:—

Name of ship.	Length in feet.	No. of shafts.	Beam as percentage of length.	Midship section coef.	Block coef.	Prismatic coef.	Appendage resistance in percentage of bare hull resistance.
Chester . .	420	4	11·2	·724	·400	·553	11·3
Columbia . .	411·58	3	14·1	·869	·491	·566	13·2
50-ft. launch .	50	1	20·0	...	·352	...	2·7
Fuel barge .	160	1	15·6	·980	·886	·904	3·6
Sonona . .	175	1	19·5	·875	·531	·607	3·4
T.B. Mackenzie	99·25	1	12·9	·700	·420	·600	2·3
T.B.D. Smith .	289	3	9·0	·649	·407	·628	9·7
T.B. Talbot .	99·5	1	12·6	·800	·337	·421	3·6
Utah . .	510	4	17·3	·9792	·5837	·596	15·8
Vicksburg .	168	1	21·4	·820	·482	·589	3·0
Wyoming .	554	4	16·8	·986	·618	·628	15·4

The percentage additions for appendage resistance for full-sized ships, given in Captain Dyson's book, were based upon experi-

ments upon models with and without the appendages. While 2½ to 3½ per cent. is about correct for single-screw ships, the appendage resistance for some two-, three-, and four-screw ships is apt to be exaggerated when deduced by this method from models.

Mr Luke's experiments with a model twin-screw ship of ·65 block coefficient, and ratio of length to beam = 6·8, quoted in Mr Baker's book, showed the resistance varying with angle of bossing, thus :—

Angle of bossing to horizontal.	0°.	22½°.	45°.	67½°.
Percentage addition for bossing and webs over and above the resistance of naked model	9·7	4	2·6	5

Note the high resistance of the horizontal bossings compared with those sloped normal to the hull. Mr G. S. Baker, in his Newcastle lecture, 1915, mentioned the uselessness of attempting to ascertain the resistance of full-sized brackets or bossings from small-scale experiments.

For building up the calculated total E.H.P. from the naked model, it may be remembered that associated with horizontal bossings there is a high hull efficiency value with outward-turning screws, and if we must assume something, we may perhaps say 9 per cent. with ordinary merchant twin-screw shaft bossings, and 7 per cent. with A brackets ; for three-screw ships about 10 per cent., and four-screw ships 9 per cent. increase for appendage resistance.

I. The Cost in Power of Bilge Keels.

In a paper read before the *American Society of Naval Architects and Marine Engineers* in 1914, Professor C. H. Peabody gave results of elaborate tests carried out on the self-propelled experimental vessel "Fulton," 30·9 ft. in length, the keels being about 15 ft. in length. The bilge keels used would have been 7½ ins. thick and from 30 ins. to 7 ft. 6 ins. in depth for a similar ship 309 ft. in length, instead of being, as generally made, viz. with a single bulb-plate of practically negligible thickness and two angles to shell of ship. As remarked by *The Engineer*, 6th February 1914, in an excellent article, the amount obtained from these experiments for added resistance may be looked upon with a certain amount of doubt as a measure of that required for normal keels.

Added Resistance due to Skin Friction.—In Wm. Froude's experiments on the "Greyhound" the added resistance when the ship fitted with bilge keels was towed was said to be less than that computed from surface friction alone. Whether the bottom of the ship was slightly cleaner or not when the bilge keels were tried we do not know. "The surface-friction calculation is based on the assumption that the forward end of the bilge keels in their advance meet with undisturbed water, while, as a matter of fact, the bilge keels being situated at the middle of the ship are not meeting undisturbed water, but water that has already been put in forward motion by the bow of the advancing vessel; that is, they are to some extent in the frictional wake, and this would reduce the actual surface-friction resistance below that computed."*

Added Resistance due to Eddying.—Taylor † points out that model experiments show that when bilge keels follow the lines of flow and are sharpened at the ends, the additional resistance due to them is not greater than that due to the additional surface alone, and that they may be placed at appreciable angles to the natural lines of flow without greatly augmenting resistance beyond that due to their surface, there being but little eddying around model bilge keels, whereas with full-sized ships if the bilge keels do not follow the lines of flow there may be a great deal of eddying.

[It has been stated that the small power for a gyro is only required when the necessity for stabilising arises, while bilge keels are a drag in all weathers.]

II. Appendages.

The resistance of appendages, viz. bossings, ram (if any), immersed counter (if any), bilge keels, sometimes docking keels, rudder, shafting, shaft struts, propeller bosses, spectacle frames, is chiefly eddy resistance, and may be minimised by careful shaping. With single-screw vessels it may amount to 4 per cent. of the resistance of the naked hull, and with twin-screw ships, according to Mr Taylor, it may be as great as 20 per cent., though usually much lower than this, often about 9 per cent. Long cones materially assist in reducing the resistance of propeller bosses, which, if large in diameter, do not greatly affect appendage resistance when the propellers are slow-running. When the propellers are fast-running, then solid propellers with small hubs are preferable from the point of view of resistance

* *Ibid.*

† *Speed and Power of Ships*, by D. W. Taylor (Chapman & Hall, 1911), p. 123.

of appendage. The angle of the web of the spectacle frame or shaft boss may advantageously be placed edgewise to the flow of the stream lines, in what Mr D. W. Taylor calls the neutral position.

As a matter of fact, in a great many moderately full merchant ships the resistance is greater than it need be, either because too little attention is paid to this angle, or because it is cheaper to build nearly horizontal. To keep the web in the neutral position the angle would have to vary along the length of the shaft boss.

Powering Ships.

The resistance of the naked model is the basis upon which the power for the full-sized ship is estimated. From the E.H.P. curve of the naked model the propulsive coefficient is obtained, viz. $\frac{\text{E.H.P. (naked)}}{\text{I.H.P.}}$ or $\frac{\text{E.H P. (naked)}}{\text{D.H P.}}$.

From Taylor's contours an estimate of this E.H.P. can be approximately obtained, and, divided by the I.H.P. or S.H.P.; gives what Rear-Admiral Taylor calls "a nominal efficiency of propulsion." Taylor's contours may be used for calculating the E.H P. (naked) and for checking results from models.

Unless, however, the air resistance and the appendage resistance are added to the E.H.P. (naked) from model or from Taylor's contours, and a new E.H.P. taken as the numerator in the fraction $\frac{\text{E.H.P.}}{\text{Hull efficiency}}$, we do not obtain a large enough T.H.P. to start with for propeller calculations. The appendage resistance is not a factor in the effect of propeller action on the resistance of the hull—it is a larger percentage than anything we can charge to mere propeller action. Similarly air resistance should not be included in propeller efficiency—it should be part of the gross E.H.P.

Therefore we write

$$\frac{\text{E.H.P. (naked)} + \text{air H.P.} + \text{appendage H.P.}}{\text{D.H.P.}} = \text{propulsive efficiency},$$

and

$$\frac{\text{E.H.P. (naked)} + \text{air H.P.} + \text{appendage H.P.}}{\text{Hull efficiency}} = \frac{\text{Gross E.H.P.}}{\text{Hull efficiency}} = \text{T.H.P.}$$

and

$$\text{Screw efficiency } e_2 = \frac{\text{T.H.P.}}{\text{D.H.P.}} = \frac{\text{Gross E.H.P.}}{\text{D.H.P.}} \times \frac{1}{\text{Hull efficiency}}.$$

Taking $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ at the figure usually quoted, viz. ·50, or sometimes ·55, *i.e.* the E.H.P. deduced from the naked model in the tank, the figure should be multiplied by about 2, to give the I.H.P. for the ship at the corresponding speed. This multiple is often assumed a sufficient guide for enabling the builder to predict the performance for a measured mile trial, or even for a run, say, from the Clyde to Liverpool. Under sea-going conditions, however, after the vessel is commissioned, a speed lower by $\frac{3}{4}$ knot to $1\frac{1}{4}$ knot than the trial-trip top speed is all that is expected and obtained. Obviously, then, coefficients of performance, obtained from vessels driven on trial at speeds corresponding to their forms, have to be modified not a little in some more or less rough way before applying the same methods to "sea speeds." It is necessary to take account of the meteorological conditions prevailing on given ocean-trade routes. On her voyage the ship encounters (1) waves which not only affect resistance by temporarily altering the trim, but which have to be reversed in direction of motion before the wave-making proper to the ship's motion can be developed; (2) ocean currents, which alter the actual speed of the ship, and which should be provided for; and (3) air resistance from prevailing winds and other winds.

Such information, obtainable from Meteorological Survey records, can be tabulated for the use of the engineer. It should be possible to translate these items of information into percentage factors directly affecting ship resistance, so that the difference between trial speed and sea speed may be estimated if not calculated, instead of guessed.

The special conditions of the service on each trade route are known and understood by the staffs of the shipowners concerned, and this partly explains why text-books are so little used by them. In determining the most suitable proportions for the propeller, these considerations are even more cogent, for, though the figures calculated in accordance with the most learned monographs may give results, in smooth-water measured mile trials, beyond the contract requirements, they frequently fail to produce the propellers which are needed for thrashing along at the necessary speed at sea to gain a tide, even when the trial-trip speed specified in the contract is the usual knot, or knot and a half, more than the required sea-speed.

(1) The skin-frictional resistance of the wetted surface of ship can be calculated (see Skin Friction, p. (9), and a percentage addition given in Mr Baker's way to this resistance, to allow for increased resistance due to the form, and another percentage added for rough

bottom, and a certain percentage for skin friction of appendages, such as rudder, bilge keels, propeller struts and shaft bossings, and for deadwood when this is in excess of the usual amount.

(2) The wave-making resistance which follows from the propagation of diverging waves from the bow and stern and transverse waves from the immersed hull may be closely estimated from model experiments, or from Mr Taylor's contours of residuary resistance per ton of displacement, with the necessary modifications for parallel body if required; and percentage additions may be given for the influence of changes of trim, rolling and pitching involving retardations, rough water tending to disturb the regular formation of waves, rolling and pitching placing the ship in positions which cause the total resistance to be increased.

(3) The eddy-making resistance, the equivalent of energy imparted to the water in churning it into eddies, due to irregular motion of rudder, and to the irregular closing of the water round blunt-ended appendages such as propeller struts and webs, and broken water round the stern-post, stem, and bilge keels, may be roughly estimated. (1), (2), and (3) together constitute the total water resistance, the gross tow-rope resistance. The useful work performed in overcoming these three is E.H.P.

(4) The augmentation of resistance occasioned by the presence and action of the propellers. The useful work performed in overcoming (1), (2), (3) and (4) is T.H.P., the H.P. delivered by the screws in propelling the naked hull without air resistance.

(5) The air resistance, affected by differences in the force of the wind, may be estimated approximately from the formula $R = KAV^2$, where R is the air resistance in lbs. of a plane area A in square feet of the transverse above-water projection of the ship, including funnel, etc., moving normally to the direction of motion of the vessel at a speed V in knots, and $K =$ a constant given by Admiral Taylor as ·0035 to ·005. The horse-power absorbed in overcoming $R = \dfrac{R \times V \times 101 \cdot 33}{33\,000}$.

(6) The appendage resistance is included in (1), (2), and (3).

If we call the horse-power delivered to the propeller the D.H.P., then $\dfrac{\text{Work got out}}{\text{Work put in}} =$ efficiency, we have $\dfrac{\text{T.H.P.}}{\text{D.H.P.}} =$ propeller efficiency. $\dfrac{\text{D.H.P.}}{\text{S.H.P.}} =$ shaft transmission efficiency, and $\dfrac{\text{S.H.P.}}{\text{I.H.P.}} =$ engine efficiency. The friction of the propelling machinery represented by I.H.P. – D.H.P. may be estimated, or taken from Plate 41.

Ship.	L × B × D.	Δ	Block coef.	Speed.	I.H.P. each screw.	Propeller H.P.	Revs.	App. slip. per cent.	Propeller dia.	Face pitch.	Exp. surface.	Exp. area ratio.	Nominal pitch ratio.	No. of propellers.	No. of blades.	Propeller.	Propeller efficiency.	Real slip ratio (face pitch).	Thrust in lbs.	"A."	$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$
Denholm Young's A 1915 tank steamer	380 × 52·75 × 23·5	10 780	·80	10·47	1 628	1 335	70	1·13	17·5	15·0	85	·354	·86	1	4	C.I. solid	·624 8	·364 5	25 900	2·4[illegible]	343
Denholm Young's B 1915 tank steamer	372 × 50·75 × 24·83	10 400	·776	11·25	2 350	1 927	73	5·5	17·75	16·5	100	·40	·90	1	4	,,	·623 5	·375	34 730	2·675	289
Denholm Young's E 1915 tank steamer. Sea performance	294 × 38 × 23	5 500	·75	9·6	1 095	898	75	16·5	13·75	15·75	75	·50	1·15	1	4	,,	·588 5	·432 5	17 900	3·1	252
S.S. T. (A) . .	260 × 35·18 × 17·25	3 533	·784	9·695	864	709	61·5	12·4	14·0	18·25	58	·376	1·304	1	4	,,	·594 5	·423 7	14 160	3·55	234
S.S. M. (A) .	275 × 36·08 × 17·46	3 670	·74	11·34	1 326	1 087	89	7·78	14·0	13·0	61	·396	·929	1	4	,,	·620 8	·395 8	19 400	2·392	262
S.S. T. (L). At sea	440 × 51·35 × 26	13 925	·785	10·9	2 709	2 220	62	9·65	18·75	19·75	103·5	·374	1·053	1	4	Built	·615 1	·406	40 800	3·26	277
T.S.S. H_2 . .	440·3 × 54·1 × 23·5	10 195	·637	14·75	$\frac{5\,600}{2}$	..	85	6·22	16·75	18·75	72	·326	1·12	2	3	Bronze blades	·718	·200 5	38 100	2·94	270
Cargo steamer .	450 × 55 × 27	14 920	·782	11·0	$\frac{2\,934}{2}$	1 231	75·5	..	14·75	17·5	58	·39	1·19	2	3	,,	..	..	..	..	275
Scout Salem, U.S.N.	420 × 46·75 × 16·8	3 750	·407	25·0	$\frac{16\,576}{2}$ S.H.P.	..	357	..	9·5	8·66	43·7	·616	·912	2	3	Bronze	..	..	..	..	237
S.S. A. . .	400·4 × 50·1 × 23·5	9 170	·68	14	3 900	3 225	75	7·8	19·0	20·5	100	·352	1·08	1	4	Bronze blades	·700 3	·345 5	52 600	2·3	308
S.S. P. (at sea) (design)	340 × 46·5 × 23·33	8 000	·76	10·5	1 950	1 640	70	11·5	16·75	16·75	91	·412	1·0	1	4	C.I. solid	..	..	..	..	238

The propeller losses constitute a gap not easily filled by calculation, but we recommend Mr R. E. Froude's 1908 efficiency curves, and the method of using them adopted in our propeller calculations.

Japanese battleship "Kongo," 1913, built at Barrow, four screws, Parsons turbines direct. Yarrow large-tube boilers, 275 lbs. W.P. Length over all = 704 ft. Length b.p. = 653. Water-line = 692 ft. Beam = 92 ft. Designed draught = 27·5 ft. Displacement at designed draught = 27 500. Block coef. = ·55. Designed speed = 27·5 knots. Designed power = 64 000 S.H.P. Propellers = 12 ft. dia. $\frac{V}{\sqrt{L}} = 1{\cdot}045$.

Hull equipment and stores . . .	13 400 tons
Armament and ammunition . . .	4 000 ,,
Armour	4 500 ,,
Propelling machinery	4 500 ,,
Coal	1 100 ,,
Total =	27 500 tons

DESTROYERS.

	Argentine "Jujuy."	Chilian "Almirante Lynch."	U.S. 1911 programme.
Length W.L. .	286′ 6″	320′	300′
Beam . .	27′	32′ 6″	30′ 3″
Trial draught .	8′ 8½″	9′ 10″	9′ 3″
Trial displac. (tons)	995	1 560	1 010
No. of screws .	2	3	2
Machinery .	Germania turbines direct	Parsons turbines direct	Parsons turbines direct
Contract speed, knots (6 hours' trial)	32	31	29
S.H.P. . .	24 000	29 000	16 000
Block coef. .	·52	·534	·421
$\frac{\Delta^{\frac{2}{3}}V^3}{\text{S.H.P.}}$ . .	136	138	153
$\frac{V}{\sqrt{L}}$ . . .	1·892	1·735	1·678

Approximate apportionment of weights in 1 000-ton destroyer.

Steel hull	285 tons
Woodwork	10 „
Fittings	65 „
Propelling machinery	482 „
Armament	48 „
Fuel and stores	92 „
Margin	18 „
	1 000 tons

The stern lines are round. The forward lines are almost straight (very slightly hollowed). The bottom of the hull begins to rise from the base line at a point on the keel about 19 per cent. of the length of the vessel measured from the aft end of the immersed hull. The post of the all-under-water-type rudder is about 15 ft. from the aft end of the immersed hull.

A typical torpedo-boat destroyer of 1911–12 has midship section coefficient of ·825, with $\frac{\text{Beam}}{\text{Draught}} = 3{\cdot}83$. The lines are shown in an article by Mr W. Lambert in *The Shipbuilder*, December 1913.

The weights are mentioned as being approximately apportioned as follows:—

Steel hull	285 tons
Woodwork	10 „
Fittings	65 „
Propelling machinery	482 „
Armament	48 „
Fuel and stores	92 „
Margin	18 „
	1 000 tons

The following particulars are noted from a paper by Sir Alexander Gracie to the Institution of Civil Engineers in 1913:—

Cargo Steamers. (A voyage of 3 000 miles.)

Length.	Speed in knots.	Weight of vessel in tons.	Tons weight of cargo.	Coal consumption in tons for the voyage.	Tons coal consumed per voyage per 100 tons cargo.	Tons weight of constructive material per 100 tons of cargo when ship is fully loaded.
400	13	3 700	4 000	500	12½	92½
500	13	6 750	8 700	700	8	77½

Turbine-driven Channel steamer "Newhaven," built in 1910. $292 \times 34 \cdot 6$ ft. beam.

Triple screw, three direct turbines. Water-tube boilers. Trial speed 23·85 knots. $\frac{V}{\sqrt{L}} = 1 \cdot 4$. 1 510 tons displacement. 13 000 S.H.P. from a weight of machinery of 590 tons, or 22 S.H.P. per ton of machinery, being $2\frac{1}{2}$ times that obtainable from paddle machinery and double the output of twin-screw reciprocating engines.

Channel steamer "Ibex." Date 1891. $265 \times 32\frac{1}{2} \times 15\frac{1}{2}$. 1 062 tons gross. 4 200 I.H.P. 19·37 knots. $\frac{V}{\sqrt{L}} = 1 \cdot 19$. Twin-screw reciprocating (three-cylinder triple) machinery developed $10\frac{1}{4}$ I.H.P. per ton.

Weights:—

Hull	60 per cent.
Machinery	30 ,,
Coal	$4\frac{1}{2}$,,
Passengers, stores and water . .	$5\frac{1}{2}$,,
	100 per cent.

Paddle Channel steamer "Calais-Douvres." Date 1893. $324 \times 36 \times 14$. 1 065 tons gross. Unclassed. 6 000 I.H.P. 20·64 knots. $\frac{V}{\sqrt{L}} = 1 \cdot 15$.

The hull weighed .	805 tons.	
Machinery . . .	650 ,,	($9\frac{1}{4}$ I.H.P. per ton).
Coal	103 ,,	

Hull	48 per cent.
Machinery	39 ,,
Coal	6 ,,
Passengers, stores, and water . . .	7 ,,
	100 per cent.

Old Cunarder T.S.S. "Campania." Built in 1893. $600 \times 65 \times$ 41 ft. 6 ins. 13 000 tons gross. 22 knots at sea. 30 000 I.H.P. 480 tons coal per day. Triple-expansion engines, 165 lbs. pressure. $\frac{\text{Length}}{\text{Depth}} = 14 \cdot 45$. 69-in. stroke.

Weight of hull	$48\frac{1}{2}$	per cent. of the	displacement.
" machinery	$21\frac{1}{2}$	"	"
" fuel	$14\frac{1}{2}$	"	"
" passengers, stores, and water	$4\frac{1}{2}$	"	"
" cargo	11	"	"
	100	per cent.	

Consumption $1\frac{1}{2}$ lbs. per I.H.P. hour.

"Adriatic." Registered dimensions :—709·2 × 75·5 × 56. Built in 1906. Twin-screw quadruple expansion engines of about 15 000 I.H.P. 15 knots. 2 500 tons coal. 6 500 tons cargo.

Of her displacement :—

Hull	56	per cent.
Machinery	10	"
Fuel	8	"
Cargo	21	"
Passengers, stores, and water	5	"
	100	per cent.

$9\frac{1}{2}$-knot cargo steamer with poop, bridge, and forecastle. Poop = 21 ft. Bridge = 90 ft. Forecastle = 32 ft. 325 ft. 0 in. × 47 ft. $11\frac{1}{2}$ in. × 20 ft. 6 in. draught. 7 284 tons displacement. Deadweight = 4 878 tons. Bunkers = 400 tons. Machinery = 330 tons. Total invoiced materials = 1 574 tons. Outfit and remainder = 241 tons.

[From Mr John Ward's presidential address, Inst. Engineers and Shipbuilders, Scotland, 1907 :—

Weight of hull and fittings	10 610	tons.
" engines, boilers, and water	4 625	"
" fuel carried	3 163	"
" cargo	1 052	"
Displacement loaded	19 450	"]

The following particulars, giving comparative weights, etc., of direct and geared turbines, quadruple reciprocator sets and three-screw combination sets, received from shipbuilders, were published by *The Syren*, 1st July 1914 :—

	Vessel 600 ft. × 76 ft. × 26 ft. draught. Displacement = 22 600 tons. Designed speed on trial = 19½ knots.			Vessel 480 ft. × 58 ft. × 28 ft. draught. Displacement = 17 170 tons. Designed speed = 14½ knots on trial.	
	Direct turbines.	Geared turbines. Two screws.	Combination. Three screws.	Quadruple. Two screws.	Geared turbine. Two screws.
Horse-power	20 950	19 900	22 250	7 000	7 000
Total weight of machinery in tons	30 60	2 910	4 390	1 345	1 050
Tons coal per hour	13·6	12·5	14·4	4·8	4·1
Length and breadth of engine-room	68′ × 76′	44′ × 76′ + 21′ × 31′	78′ × 76′	33¾′ × 58′	33¾′ × 58′
Length and breadth of boiler-rooms	160′ × 40′	148′ × 40′ + 12′ × 20′	160′ × 40′	56¼′ × 37½′	56¼′ × 35¼′
$\dfrac{\Delta^{\frac{2}{3}} V^3}{\text{Power}}$	284	298	266	276	276

Shaft horse-power is given for turbines, indicated horse-power for reciprocating engines, and shaft horse-power and indicated horse-power combined for the combination arrangement of reciprocating engines on wing shafts and direct turbine on centre shaft.

Steam superheated 200° F. has improved all the above from a coal consumption point of view. The boilers have diminished in bulk slightly. Combination sets have given place to double-reduction geared turbines with superheated steam in the larger ships. Mr Dornan states that the combination three-screw scheme with 6 per cent. to 7½ per cent. less steam consumption than two-screw quadruple reciprocating saturated, would lose about 4½ per cent. by lower hull and propeller efficiencies, and perhaps 3 per cent. in commercial value through larger engine-room and decreased deadweight.

Twin-screw turbine Channel steamer "Konigin Luise" (see Professor Sir J. H. Biles's report, dated 1914). 275 ft. b.p. × 38·7 × 9·75 ft. load draught. Δ = 1 800. Yarrow boilers. Howden's F.D. 70° superheat. Superheating surface = 3 000 sq. ft. 240 lbs. per sq. in. Total boiler heating surface = 12 220 sq. ft.

Grate = 258·1 sq. ft. Each turbine set 3 000 b.h.p. at 1 800 revs. per min. Astern power 70 per cent. of ahead power. Fottinger transformer, with reduction ratio 4 : 1 at full power. Efficiency 88 per cent. to 89 per cent. 20 knots on trial, with 5 330 S.H.P. on 453 revolutions of propellers. 12 lbs. steam per S.H.P. hour, which compares favourably with the 15·1 lbs. of the direct-driven turbine steamer "Cæsarea" at 6 675 S.H.P. Coal analysis: moisture 2·7 per cent., ash 9·41 per cent., volatile 11·85 per cent., sulphur 0·70 per cent. Calorific value 12 220 B.Th.U. Consumption 6 321 lbs. per hour, or 1·38 lbs. per S.H.P. hour, on three hours' full-power trial. Propellers: diameter = 6 ft. 6¾ in.; pitch = 5 ft. 7 in.; projection area = 18·1 sq. ft. Developed area = 20 sq. ft.

One Curtis Vulcan combined impulse and reaction turbine on each shaft. 176 lbs. pressure in receiver. Weight of turbines and gearing = 42 tons. Professor Sir J. H. Biles's estimate of steam consumption for auxiliaries is 1·6 lbs. per S.H.P. of main turbines, *i.e.* 12 + 1·6 = 13·6 lbs. steam per S.H.P. hour total. The 12 lbs. were actually measured. With coal of calorific value 1·31 B.Th.U. the consumption was 7 050 lbs. or 1·31 lbs. per S.H.P. hour.

Relative coal consumption for different machinery in steam cargo and semi-passenger liners. Propellers 75 to 85 revs. per min.

Machinery.	Comparison.	Coal burned.
(1) Triple-expansion reciprocating with saturated steam, 180 lbs.	Standard	100
(2) Quadruple-expansion reciprocating with saturated steam, 220 lbs.	7 per cent. gain	93
(3) Triple-expansion reciprocating with about 200° F. superheat	About 14 per cent. more economical than triple saturated	86
(4) Quadruple reciprocating with about 200° F. superheat	About 9 per cent. more economical than quadruple saturated	83¾
(5) Parsons mechanically double-geared turbines with about 200° F. superheat	About 10 per cent. more economical than quadruple reciprocating superheated	75½

In (3), (4), and (5) there is perhaps room for a very slight further reduction in coal consumption if steam superheated to, say, 50° F. is extensively used for auxiliaries, but this entails extra cost for plant and upkeep. For certain auxiliaries, such as feed water-heaters, evaporators, distillers, etc., where there are steam coils, saturated steam is required. The total steam consumption for auxiliaries is not less with turbines than with reciprocating engines because there are more auxiliaries.

The steam consumption in lbs. per I.H.P. hour for quadruple reciprocating main engines with saturated steam of 220 lbs. pressure is about $12\frac{3}{4}$, and with steam superheated 200° F. about $11\frac{1}{4}$ lbs. Direct turbines, 200 lbs. pressure, saturated, about $11\frac{1}{4}$ lbs. Direct turbines, superheated steam, about $10\frac{1}{2}$ lbs. Turbines with single-reduction gear, superheated steam, 10·1 lbs. Turbines with double-reduction gear, superheated steam, 8·4 lbs., and there are possibilities with electric gear. Turbines with hydraulic gear, superheated steam, about 10 lbs.

In a paper entitled "Some Alternative Types of Machinery for a $19\frac{1}{4}$-knot Steamer," by Mr Jas. Dornan (*Inst. Engineers and Shipbuilders, Scot.*, 1915), a comparison was made of horse-powers, efficiencies, coal consumptions, weights, etc., for seven different arrangements of engines, for an intermediate passenger and cargo type for the North Atlantic, $600 \times 72 \times 46$ of 27 ft. mean mid-Atlantic draught, $\Delta = 21\,000$ tons, $19\frac{1}{4}$ knots average at sea throughout the year. Design A is taken as a basis for comparison with the others. A = twin-screw quadruple reciprocating saturated, 85 revs. screws, 210 lbs. W.P., Howden's F.D. Steam consumption, lbs. per hour per H.P. of main engines :—

Main engines	12·71 lbs.	
Propelling machinery auxiliaries . . .	1·27 ,,	2·01.
Hull auxiliaries, make-up feed and drains	0·74 ,,	
Total . . .	14·72 lbs.	

2·01 lbs. steam per hour per H.P. of main engines is for auxiliary consumption, deck and engine, or 13·7 per cent. of the total consumption.

$$\frac{\Delta^2 V^3}{\text{I.H.P.}} = 251.$$

Ship.	Mr Dornan's A. and C.	Mr Dornan's B. and D.	Mr Dornan's E.	Mr Dornan's F.		
Design.	Two-shaft quadruple engines.	Four-shaft direct turbines	Two-shaft turbines with hydraulic gear.	Two-shaft turbines, single-gear mechanical.	Two-shaft turbines, electric gear.	Two-shaft double-reduction geared turbines.
No. of shafts	2	4	2	2	2	2
Revs. per min.	85	290	200	160	85	85
E.H.P. naked	11 430	11 100	11 430	11 430	11 430	11 430
,, with appendages . .	12 500	12 400	12 300	12 300	12 500	12 500
Wake	·165	·20	·165	·165	·165	·165
Thrust deduction	·15	·15	·15	·15	·15	·15
Efficiencies :—						
Hull	·99	1·02	·99	·99	·99	·99
Propeller model	·695	·62	·64	·64	·695	·695
,, actual	·647	·577	·595	·595	·647	647
Mechanical	·903	·97	·98	·97	...	·95
Propulsive	·577	·57	·577	·572	·609	·609
I.H.P.	21 650					
S.H.P.	...	21 800	21 300	21 350	21 350	21 350
Propulsive coef. from E.H.P. naked, *i.e.* $\frac{\text{E.H.P.}}{\text{S.H.P.}}$ or $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ . .	·528	·51	·537	·536	·536	·536

Mechanical Efficiency of Marine Oil Engines.

For four-cycle engines driving an air compressor direct, and also with circulating water and lubricating pumps attached to the engine, take ·78 as the mechanical efficiency. It may be ·80 or even ·85 in exceptional cases where the air compressor is not driven by the main motor. See pages 200 and 282.

Two-cycle engines, in which the scavenge pump, the air compressor, and the circulating water and lubricating pumps are driven by the main engine, do not usually have a mechanical efficiency much exceeding ·70.

In estimating the engine-power necessary for a Diesel-driven ship, the formula $\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}}$ may be used to begin with to find the I.H.P. which would be required if the engines were ordinary steam-reciprocating. Multiplying the I.H.P. so found by the mechanical efficiency gives the S.H.P. at the aft end of the engine, or B.H.P.

If the steamer is to run in tropical waters over 80° F. temperature (cooling water), the power of the oil engine should be increased by 10 per cent. in design work.

The weight of marine oil engines of the usual slow-speed Diesel type, including the accessories for the engine itself, is somewhere in the neighbourhood of 200 lbs. per B.H.P. for engines running between 110 and 140 or 150 revs. per min. When, perhaps in the near future, 100 revs. per min. will be usual, the weight might be relatively slightly greater, but the tendency in design will be to diminish the weight of the engines built.

The Fullager engine has the lower revolutions, better balance, and probably higher mechanical efficiency, with shorter engine room.

Results of model tests of cargo steamers with cruiser stern. 450 ft. b.p. × 58 ft. beam mld. Twin screws. A = I.H.P. with triple-expansion reciprocating steam engines under good trial conditions, no wind, clean bottom, and good design; B = I.H.P. of same, increased by 15 per cent. for sea conditions; C = S.H.P. at sea if geared turbine machinery were adopted.

Knots.	29 ft. draught. Δ = 16 000 tons. Block coef. = ·74.					29 ft. draught. Δ = 15 130 tons. Block coef. = ·70.				
	A.	B.	$\frac{\Delta^{\frac{2}{3}}V^3}{B}$.	C.	$\frac{\Delta^{\frac{2}{3}}V^3}{C}$.	A.	B.	$\frac{\Delta^{\frac{2}{3}}V^3}{B}$.	C.	$\frac{\Delta^{\frac{2}{3}}V^3}{C}$.
12	3 100	3 560	308	3 340	...	2 750	3 160	...	2 970	...
13	4 200	4 830	289	4 535	...	3 750	4 310	...	4 050	...
14	5 600	6 440	273	6 050	...	5 000	5 750	...	5 400	...
14·5	6 475	...	...	...	...	5 800	...	...	...	...

The I.H.P.'s marked A in such a table as the above may be taken as representing

$$\frac{\text{E.H.P. naked}}{\text{Hull efficiency}} + \left(\begin{array}{l}\text{Horse-power ex-}\\ \text{pended in over-}\\ \text{coming calm}\\ \text{air resistance}\end{array}\right) + \left(\begin{array}{l}\text{A percentage addition to}\\ \text{the E.H.P. naked to}\\ \text{account for the resist-}\\ \text{ance of under-water}\\ \text{appendages}\end{array}\right)$$

T.S.S. "H3". 450 ft. b.p. × 59 × draught (below). Beam as percentage of length = 13·11. $\frac{\text{Length}}{\text{Beam}} = 7{\cdot}62$. The models were made to the mean plating line. The displacements included plating, but had no allowance for any other appendage, and no allowance was made for other appendages in the results.

Model tested at the National Physical Laboratory in 1918, at three different draughts. 18 in. trim by the stern for the ship.

Mean draught in ft.	Tons Δ.	Coefficients.			$\frac{\Delta}{\left(\frac{L}{100}\right)^3}$.
		Block.	Midship section.	Mean prismatic.	
20	10 162	·676	·962	·703	111·4
23·5	12 240	·689	·968	·712	134·3
27	14 375	·702	·972	·723	157·8

RESULTS OF TANK TRIALS.

Knots.	$\frac{V}{\sqrt{L}}$.	E.H.P. from tank. 20 ft. draught.	23·5 ft. draught.	27 ft. draught.
9	·424	528	577	636
10	·471	719	784	868
11	·519	960	1 052	1 170
12	·566	1 275	1 406	1 555
13	·613	1 675	1 869	2 077
13¼	·625	1 800	2 000	2 240
13½	·636	1 930	2 147	2 408
13¾	·648	2 066	2 300	2 580
14	·66	2 196	2 480	2 746
14¼	·671	2 330	2 617	2 912
14½	·684	2 460	2 770	3 080
14¾	·695	2 600	2 925	3 250
15	·707	2 727	3 075	3 426
15¼	·719	2 862	3 230	3 612
15½	·73	3 100	3 407	3 812
15¾	·742	3 186	3 604	4 040
16	·754	3 370	3 836	4 280
16¼	·766	3 610	4 095	4 560
16½	·778	3 890	4 400	

Estimated weight of machinery with water in boilers for 6 500 S.H.P. double-reduction geared turbines, 85 revs. propeller, superheated steam 220 lbs. W.P. = 1 300 tons.

T.S.S. "H1." 418 ft. b.p. × 52 ft. × 23 ft. mean draught. 9 100 tons displacement. Block coefficient = ·637. Wetted surface = 30 100 sq. ft. Tank model tested for the Booth Steamship Co., Ltd., in 1910. Midship-area coefficient = ·956. Mean prismatic coefficient = ·666. $\frac{\Delta}{\left(\frac{L}{100}\right)^3} = 124{\cdot}6$. $\frac{\text{Length}}{\text{Beam}} = 8{\cdot}04$. Beam as percentage of length = 12·45. $\frac{\text{Beam}}{\text{Draught}} = 2{\cdot}26$.

Knots.	$\frac{V}{\sqrt{L}}$.	E.H.P. from tank.
11	·539	934
12	·587	1 240
13	·636	1 620
13¼	...	1 724
13½	·661	1 850
13¾	...	1 970
14	·685	2 089
14¼	...	2 200
14½	·71	2 325
14¾	...	2 454
15	·734 5	2 590
15¼	...	2 732
15½	·759	2 892
16	·784	3 236
16½	·808 5	3 634

Sea speed = 14·25 knots when I.H.P. = 4 600. E.H.P. naked from tank figures = 2 200. Adding 10 per cent. to 15 per cent. for sea conditions. Gross E.H.P. = 2 420 to 2 530. Propulsive efficiency = ·525 to ·55.

T.S.S. "H2." 440·3 ft. b.p. × 54·1 ft. beam × 23 ft. mean draught. Displacement = 9 912 tons. Block coefficient = ·637. Midship section coefficient = ·973. Mean prismatic coefficient = ·659. Wetted surface = 32 800 sq. ft. $\dfrac{\Delta}{\left(\dfrac{L}{100}\right)^3} = 116{\cdot}3$. $\dfrac{\text{Beam}}{\text{Draught}} = 2{\cdot}342$. $\dfrac{\text{Length}}{\text{Beam}} = 8{\cdot}14$. 14¾ knots at sea. Beam as percentage of length = 12·3. Tank model tested for the Booth Steamship Co., Ltd., in 1910.

Knots.	$\dfrac{V}{\sqrt{L}}$.	E.H.P. from tank.
11	·525	1 028
12	·573	1 335
13	·62	1 720
13¼	...	...
13½	·644	1 950
13¾	...	2 072
14	·668	2 200
14¼	...	2 338
14½	·692	2 470
14¾	...	2 670
15	·715	2 793
15¼	...	2 950
15½	·740	3 120
16	·764	3 468
16½	·788	3 900

Sea speed 14·6 knots when the I.H.P. = 5 300. E.H.P. naked from tank figures = 2 540. Adding 10 per cent. to 15 per cent. for sea conditions. Gross E.H.P. = 2 800 to 2 920. Propulsive efficiency = ·53 to ·55.

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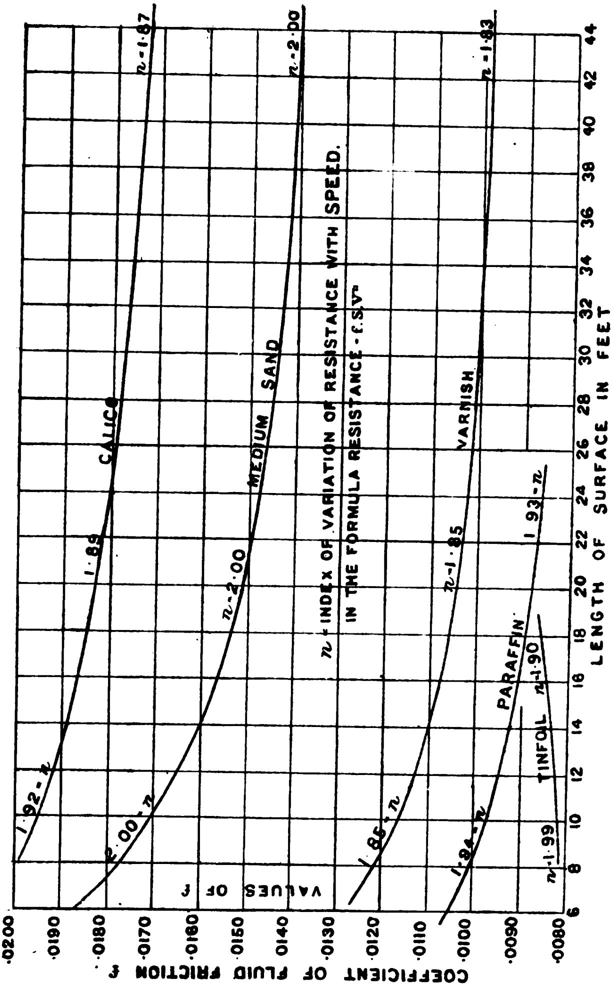
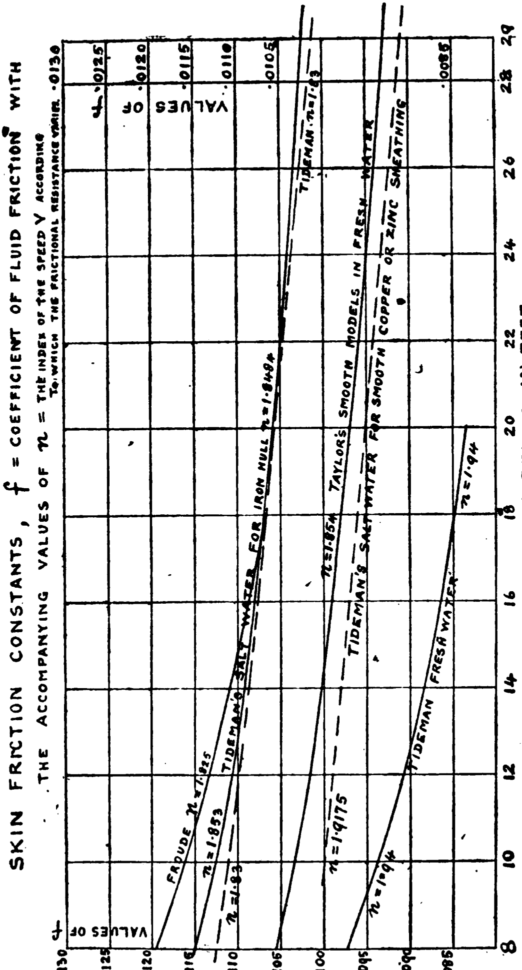


Fyfe, Steamship Coefficients.—E. & F. N. SPON, LTD.

Plate 2.



SKIN FRICTION CONSTANTS, f = COEFFICIENT OF FLUID FRICTION WITH
THE ACCOMPANYING VALUES OF n = THE INDEX OF THE SPEED V ACCORDING TO WHICH THE FRICTIONAL RESISTANCE VARIES.
VALUES OF f
·0130
·0125
·0120
·0115
·0110
·0105
·0100
·0095
·0090
·0085
FROUDE n = 1·825
n = 1·853
n = 1·83
TIDEMAN'S SALT WATER FOR IRON HULL n = 1·8484
TIDEMAN n = 1·83
n = 1·9175
n = 1·854
TAYLOR'S SMOOTH MODELS IN FRESH WATER
TIDEMAN'S SALT WATER FOR SMOOTH COPPER OR ZINC SHEATHING
n = 1·94
TIDEMAN FRESH WATER
n = 1·94
8
10
12
14
16
18
20
22
24
26
28
29
LENGTH OF SURFACES IN FEET.

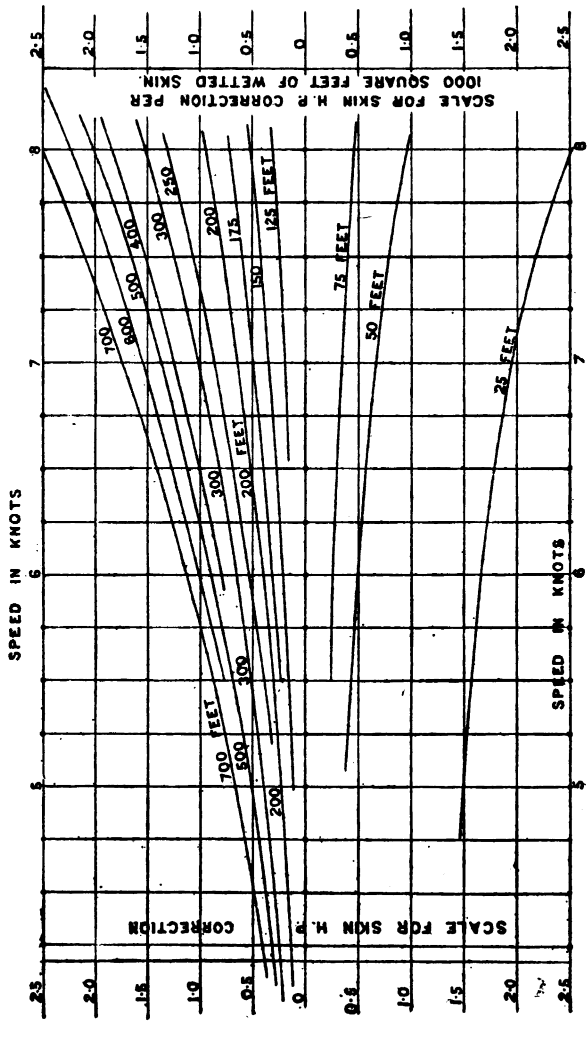

SCALE FOR SKIN H.P CORRECTION PER 1000 SQUARE FEET OF WETTED SKIN.
SCALE FOR SKIN H.P CORRECTION
SPEED IN KNOTS
SPEED IN KNOTS
700 FEET
500
300
200
125 FEET
75 FEET
50 FEET
25 FEET

PLATE 4.

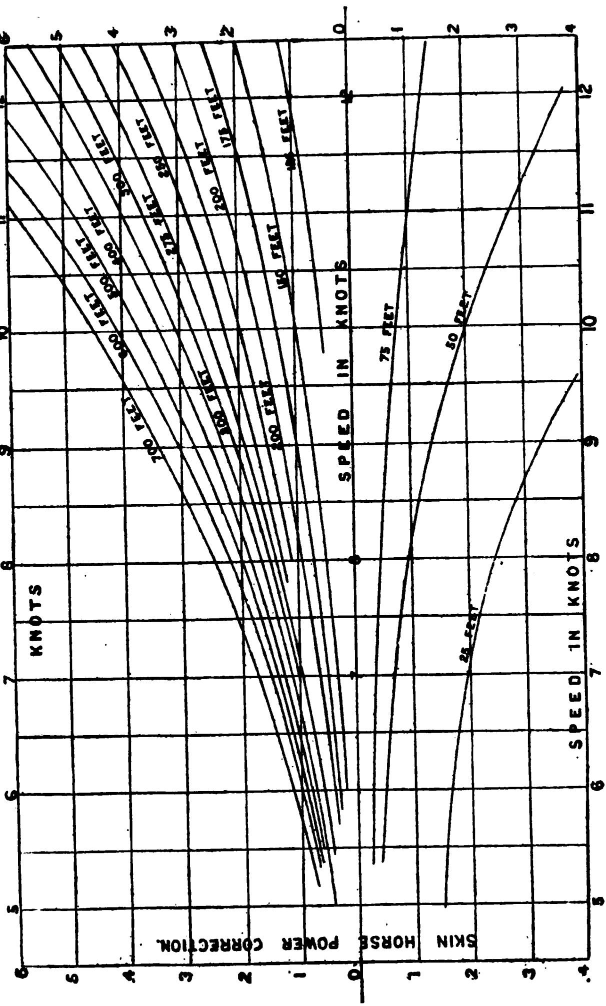
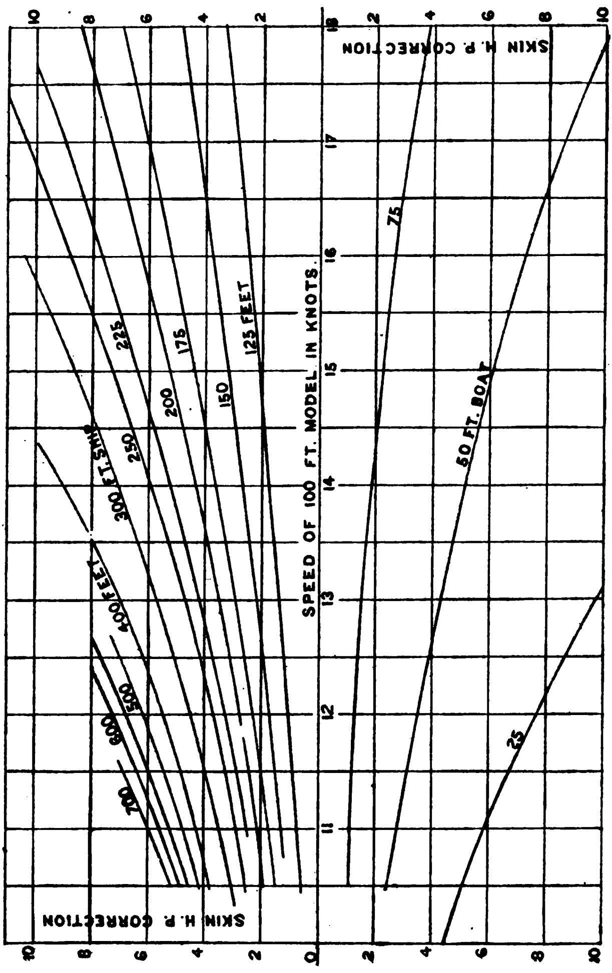

SPEED IN KNOTS
SKIN HORSE POWER CORRECTION
KNOTS
25 FEET
50 FEET
75 FEET
100 FEET
150 FEET
175 FEET
200 FEET
250 FEET
275 FEET
300 FEET
400 FEET
500 FEET
600 FEET
700 FEET

PLATE 5.


SKIN H.P. CORRECTION
SPEED OF 100 FT. MODEL IN KNOTS.
SKIN H.P. CORRECTION
11
12
13
14
15
16
17
18
10
8
6
4
2
0
2
4
6
8
10
125 FEET
150
175
200
225
250
300 FT. SHIP
400 FEET
500
600
700
75
50 FT. BOAT
25

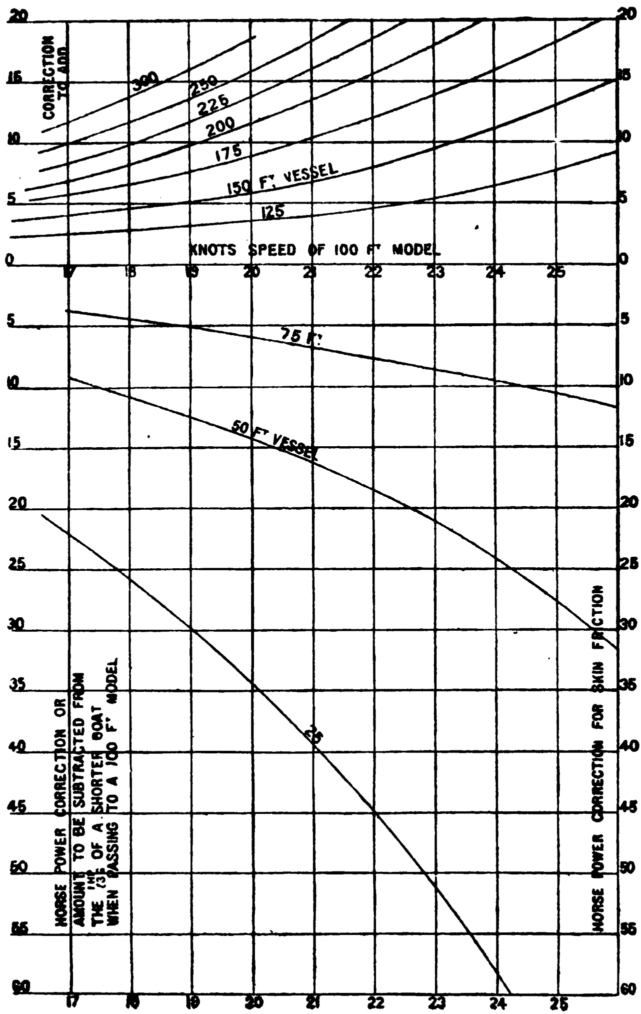
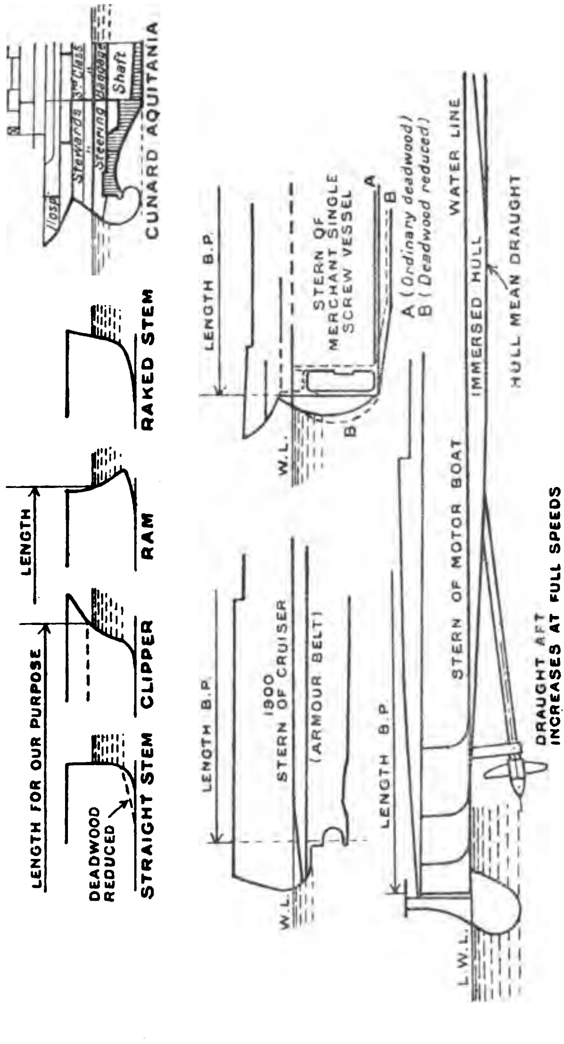
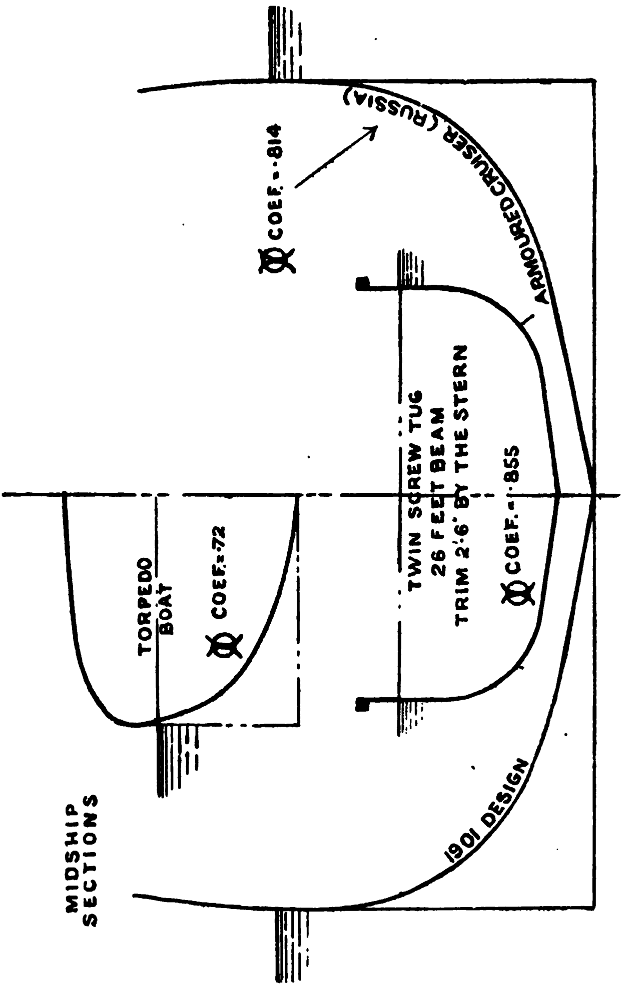
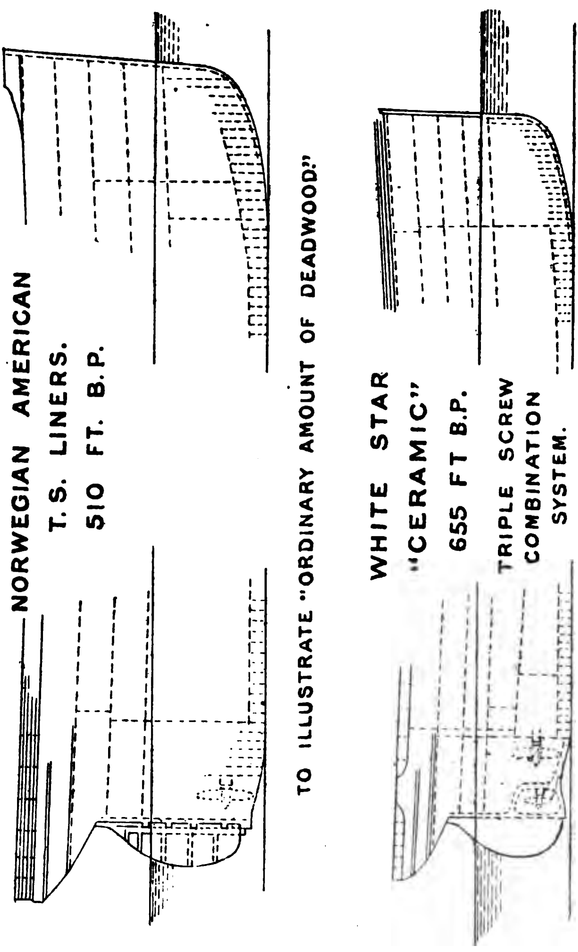

CORRECTION TO ADD
300
250
225
200
175
150 F.T VESSEL
125
KNOTS SPEED OF 100 F.T MODEL
75 F.T
50 F.T VESSEL
25
HORSE POWER CORRECTION OR AMOUNT TO BE SUBTRACTED FROM THE IHP OF A SHORTER BOAT WHEN PASSING TO A 100 F.T MODEL
HORSE POWER CORRECTION FOR SKIN FRICTION
20
15
10
5
0
5
10
15
20
25
30
35
40
45
50
55
60
17
18
19
20
21
22
23
24
25

PLATE 7.

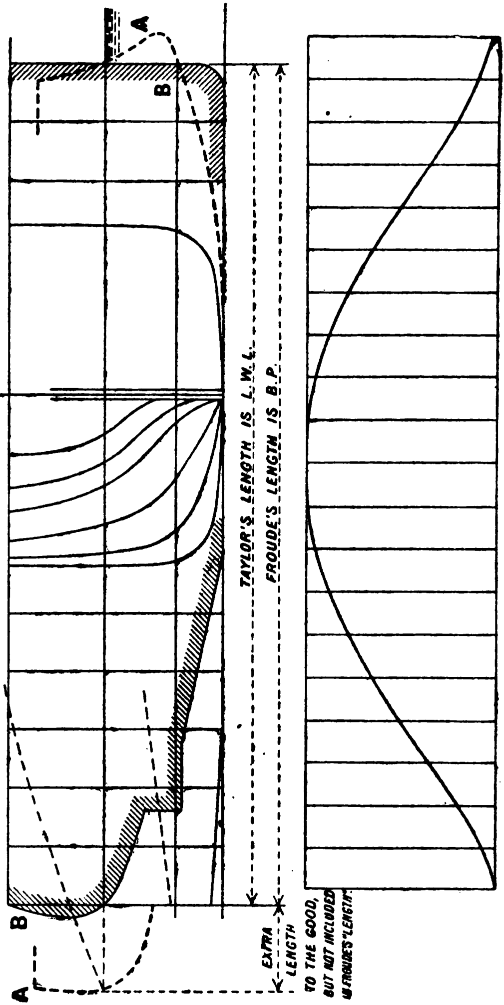

LENGTH FOR OUR PURPOSE
LENGTH
DEADWOOD REDUCED
STRAIGHT STEM
CLIPPER
RAM
RAKED STEM
CUNARD AQUITANIA
Shaft
Steering
Stewards
3rd Class
Baggage
LENGTH B.P.
1900 STERN OF CRUISER
(ARMOUR BELT)
W.L.
STERN OF MERCHANT SINGLE SCREW VESSEL
A (Ordinary deadwood)
B (Deadwood reduced)
A
B
STERN OF MOTOR BOAT
L.W.L.
WATER LINE
IMMERSED HULL
HULL MEAN DRAUGHT
DRAUGHT AFT INCREASES AT FULL SPEEDS



MIDSHIP SECTIONS
TORPEDO BOAT
COEF.=·72
TWIN SCREW TUG
26 FEET BEAM
TRIM 2'6" BY THE STERN
COEF. = ·855
ARMOURED CRUISER (RUSSIA)
COEF. = ·814
1901 DESIGN


NORWEGIAN AMERICAN
T.S. LINERS.
510 FT. B.P.
TO ILLUSTRATE "ORDINARY AMOUNT OF DEADWOOD."
WHITE STAR
"CERAMIC"
655 FT B.P.
TRIPLE SCREW
COMBINATION
SYSTEM.

A FROUDE'S TYPE 4, SERIES A, (WITH RAM STEM.)
B TAYLOR'S STANDARD SERIES, (WITH STRAIGHT STEM.)



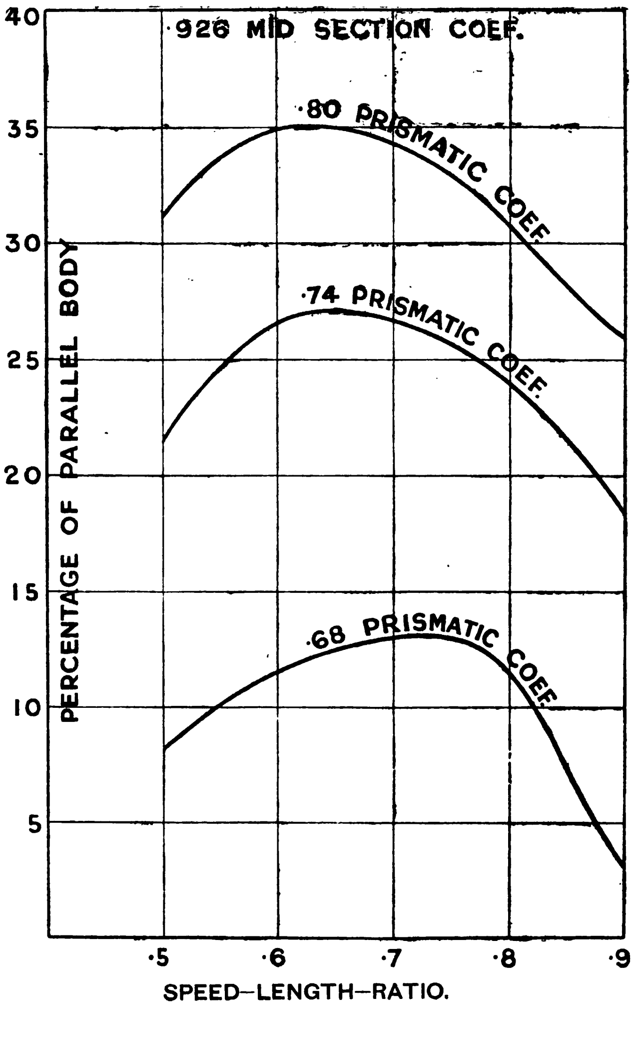

·926 MID SECTION COEF.
·80 PRISMATIC COEF.
·74 PRISMATIC COEF.
·68 PRISMATIC COEF.
PERCENTAGE OF PARALLEL BODY
40
35
30
25
20
15
10
5
·5
·6
·7
·8
·9
SPEED–LENGTH–RATIO.

PLATE 12.

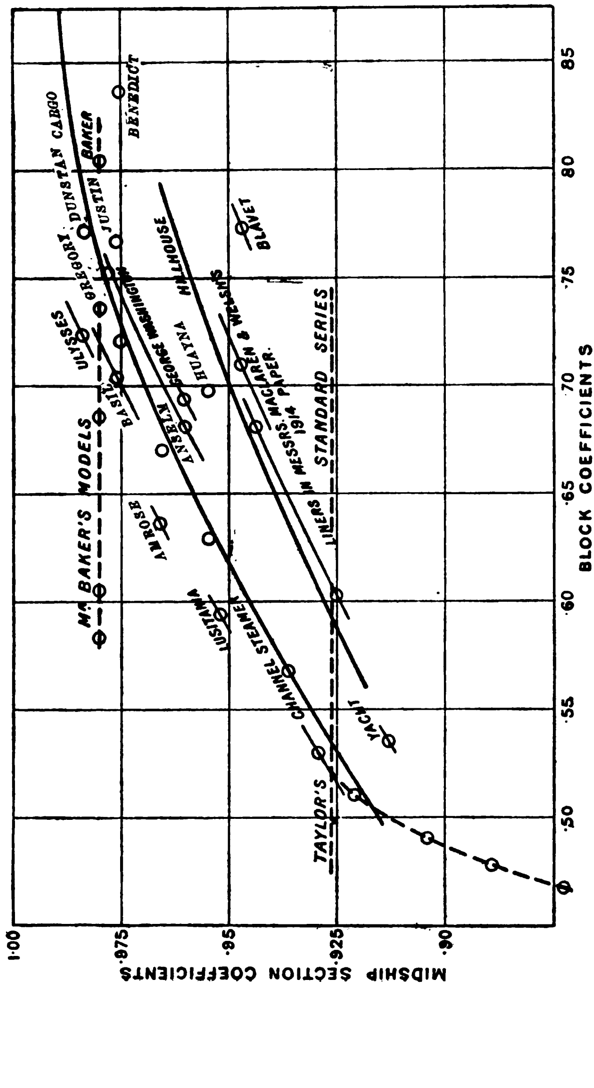
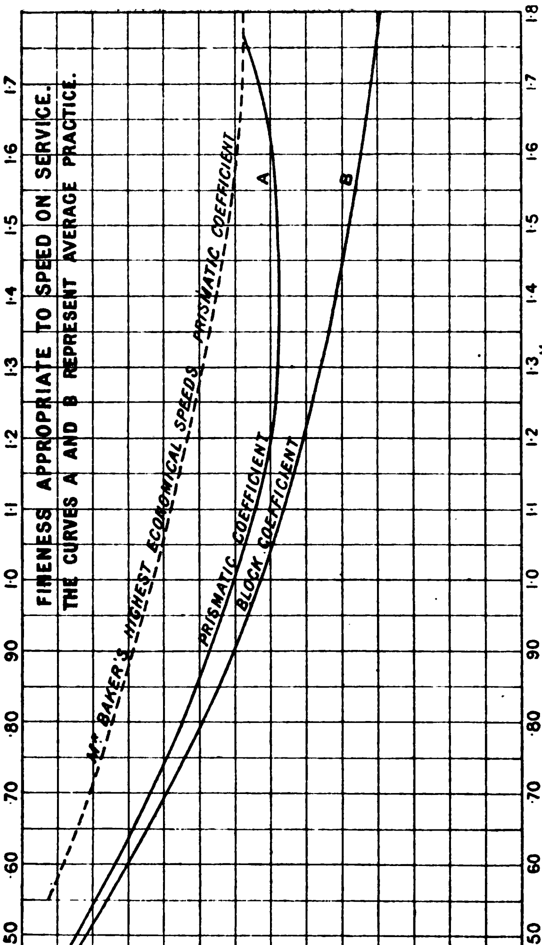

MIDSHIP SECTION COEFFICIENTS
BLOCK COEFFICIENTS
1·00
·975
·95
·925
·90
·50
·55
·60
·65
·70
·75
·80
·85
MR BAKER'S MODELS
TAYLOR'S STANDARD SERIES
LINERS IN MESSRS. MACLAREN & WELSH'S 1914 PAPER.
CHANNEL STEAMER
LUSITANIA
YACHT
AMBROSE
ANSELM
BASIL
HUAYNA
GEORGE WASHINGTON
ULYSSES
GREGORY
DUNSTAN CARGO
JUSTIN
BAKER
BENEDICT
HILLHOUSE
BLAVET

PLATE 18.



FINENESS APPROPRIATE TO SPEED ON SERVICE.
THE CURVES A AND B REPRESENT AVERAGE PRACTICE.
Mr BAKER'S HIGHEST ECONOMICAL SPEEDS — PRISMATIC COEFFICIENT
PRISMATIC COEFFICIENT
BLOCK COEFFICIENT
A
B
SPEED-LENGTH-RATIO $\frac{V}{\sqrt{L}}$.
BLOCK COEFFICIENT & PRISMATIC COEFF.
·50
·60
·70
·80
·90
1·0
1·1
1·2
1·3
1·4
1·5
1·6
1·7
1·8
·85
·80
·75
·70
·65
·60
·55
·50
·45
·40
·35

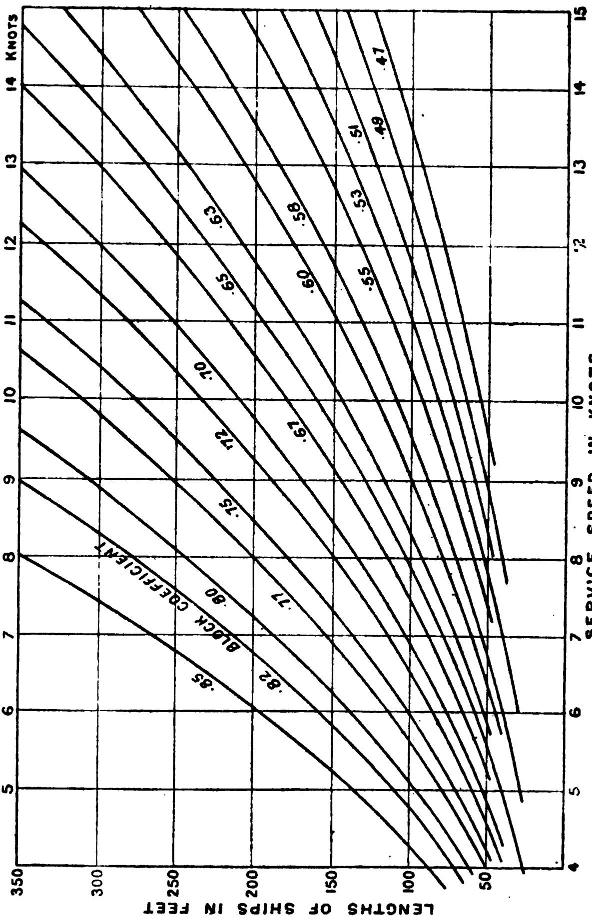

SERVICE SPEED IN KNOTS
4
5
6
7
8
9
10
11
12
13
14
15
14 KNOTS
LENGTHS OF SHIPS IN FEET
350
300
250
200
150
100
50
BLOCK COEFFICIENT
.85
.82
.80
.77
.75
.72
.70
.67
.65
.63
.60
.58
.55
.53
.51
.49
.47

PLATE 15.

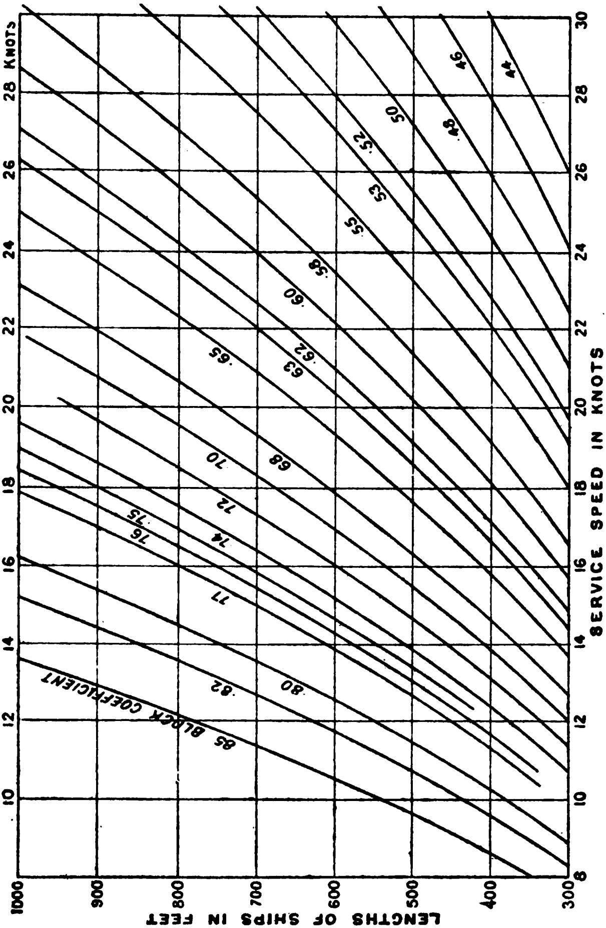
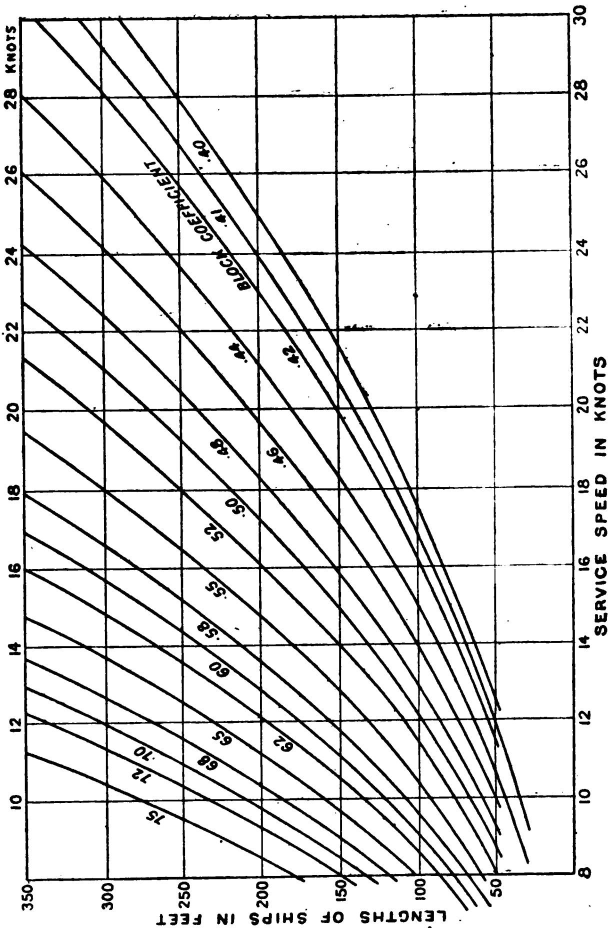
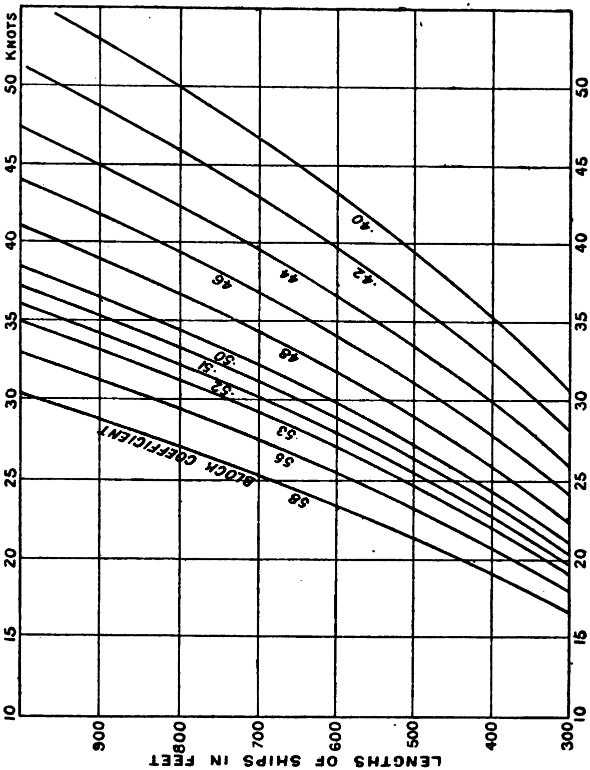

28 KNOTS
SERVICE SPEED IN KNOTS
LENGTHS OF SHIPS IN FEET
.85 BLOCK COEFFICIENT
.82
.80
.77
.76
.75
.74
.72
.70
.68
.65
.63
.62
.60
.58
.55
.53
.52
.50
.48
.46
.44
1000
900
800
700
600
500
400
300
8
10
12
14
16
18
20
22
24
26
28
30

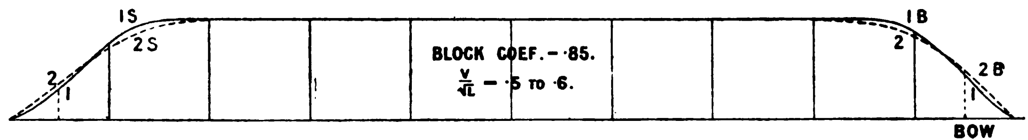
PLATE 16.


KNOTS
SERVICE SPEED IN KNOTS
LENGTHS OF SHIPS IN FEET
BLOCK COEFFICIENT
8
10
12
14
16
18
20
22
24
26
28
30
350
300
250
200
150
100
50
.40
.41
.42
.44
.46
.48
.50
.52
.55
.58
.60
.62
.65
.68
.70
.72
.75

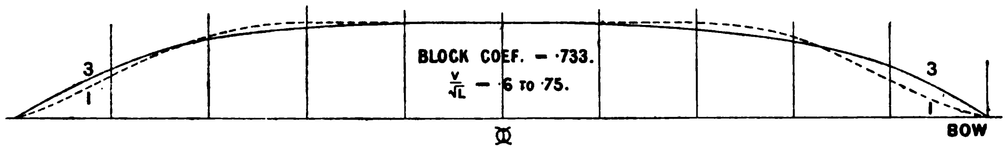


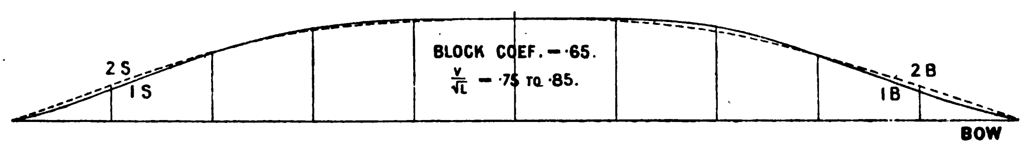
Fyfe, Steamship Coefficients.—E. & F. N. SPON, LTD.

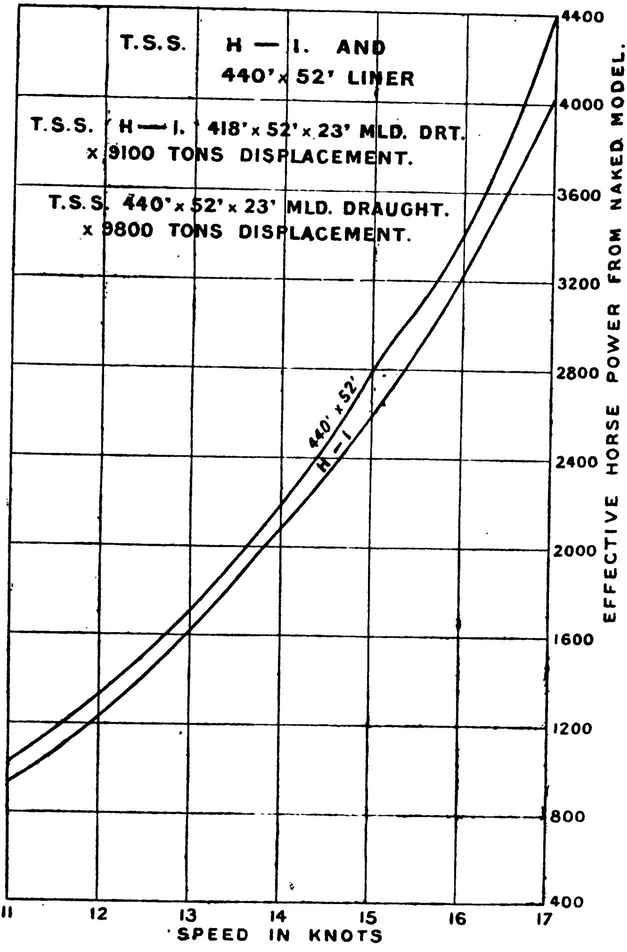
TO ILLUSTRATE PROF. SADLER'S PAPERS ON THE EFFECT OF LONGITUDINAL DISTRIBUTION OF DISPLACEMENT UPON RESISTANCE. 1907-8.

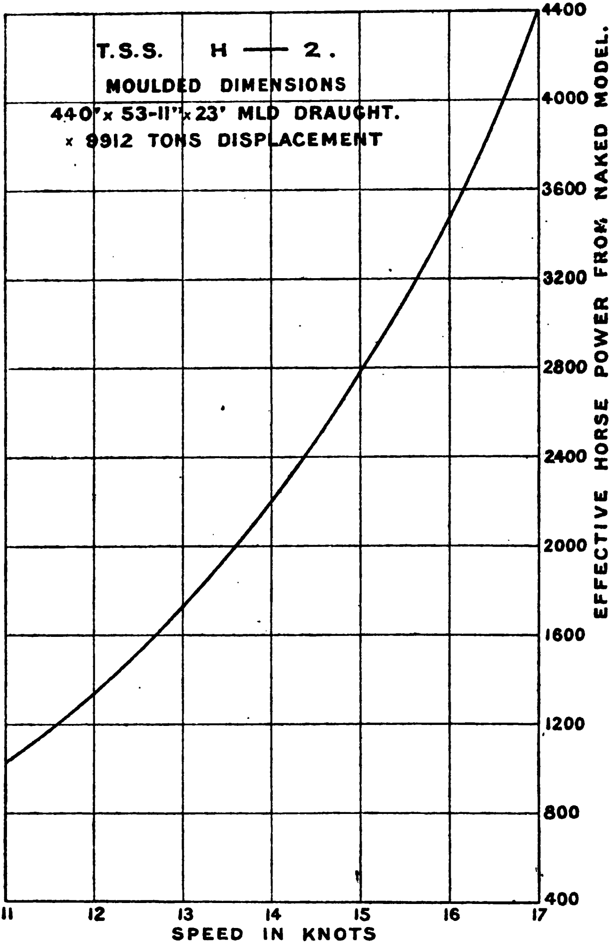


FULL DRAUGHT
SECTIONAL AREAS. F. 7.






T.S.S. H — I. AND
440' x 52' LINER
T.S.S. H — I. 418' x 52' x 23' MLD. DRT.
x 9100 TONS DISPLACEMENT.
T.S.S. 440' x 52' x 23' MLD. DRAUGHT.
x 9800 TONS DISPLACEMENT.
440' x 52'
H — I
4400
4000
3600
3200
2800
2400
2000
1600
1200
800
400
EFFECTIVE HORSE POWER FROM NAKED MODEL.
11
12
13
14
15
16
17
SPEED IN KNOTS


T.S.S. H — 2.
MOULDED DIMENSIONS
440' x 53-11" x 23' MLD DRAUGHT.
x 9912 TONS DISPLACEMENT
4400
4000
3600
3200
2800
2400
2000
1600
1200
800
400
EFFECTIVE HORSE POWER FROM NAKED MODEL.
11
12
13
14
15
16
17
SPEED IN KNOTS

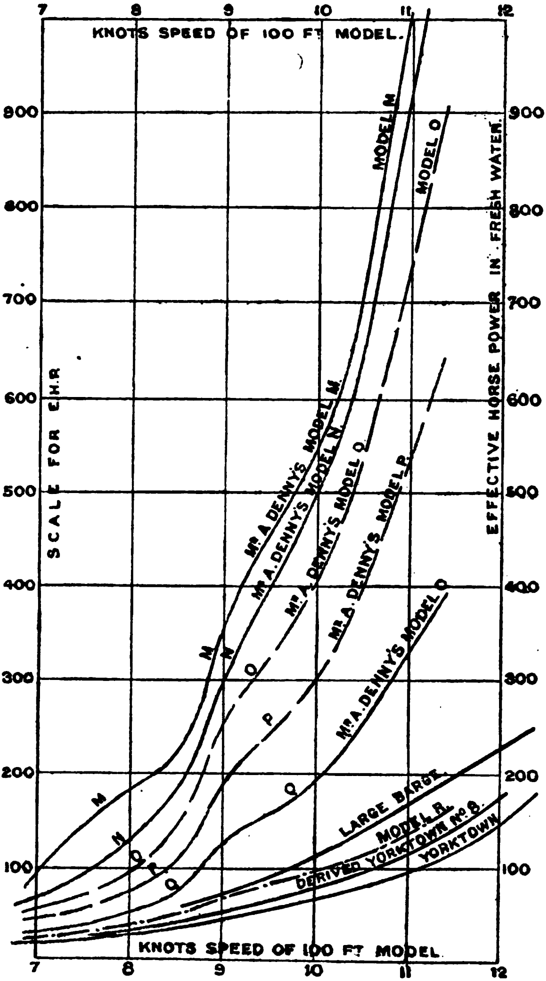

KNOTS SPEED OF 100 FT MODEL.
7
8
9
10
11
12
SCALE FOR E.H.P.
EFFECTIVE HORSE POWER IN FRESH WATER.
100
200
300
400
500
600
700
800
900
MODEL M
MODEL O
MR A. DENNY'S MODEL M.
MR A. DENNY'S MODEL N.
MR A. DENNY'S MODEL O.
MR A. DENNY'S MODEL P.
MR A. DENNY'S MODEL Q
M
N
O
P
Q
LARGE BARGE
MODEL R.
DERIVED YORKTOWN No. 8.
YORKTOWN
KNOTS SPEED OF 100 FT MODEL.

PLATE 22.

NAME	DISPLA TONS	L	B	MEAN D	COEFFICIENTS BLOCK	M/D AREA	PRISM	WET SKIN
GREYHOUND	238	100	19·25	7·98	·534	·743	·719	2532
CERAM	145·2	100	16·65	5·89	·513	·791	·656	1890
LEPANTO	230	100	18·17	7·53	·585	·884	·66	2270
MODEL R	578	100	20	1·666	·607	·833	·728	1597
YORK Nº8	138·3	100	20·8	4·57	·508			
" Nº7	138·3	100	19·58	4·68	·508			
" Nº5	138·3	100	16·97	5·62	·508			
YORKTOWN	138·3	100	15·67	6·09	·508	·843	·603	2000
YORK Nº4	138·3	100	14·36	6·65	·508			
" Nº2	138·3	100	11·75	8·14	·508			
COLORADO	108	100	13·65	4·77	·581	·972	·599	1757
MODEL A	108·2	100	11·82	6·12	·524	·924	·567	1680
ROTA Nº3	121·6	100	15·3	5·56	·50			1708
MODEL B	85·3	100	11·82	5·10	·495	·908	·545	1460
MODEL C	63·8	100	11·82	4·08	·463	·887	·522	1243
MODEL D	46·85	100	11·82	3·17	·427	·653	·501	1049

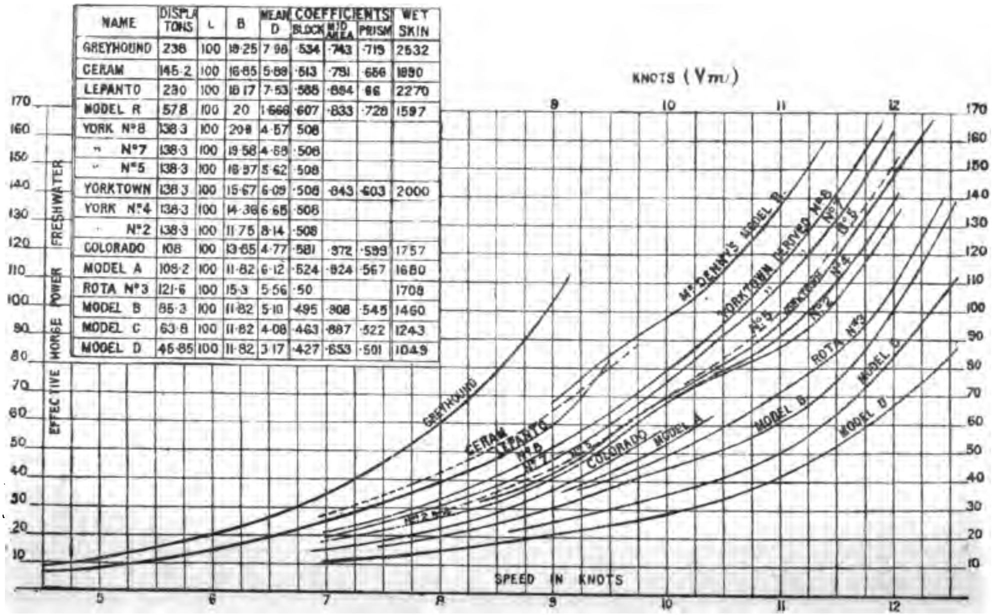
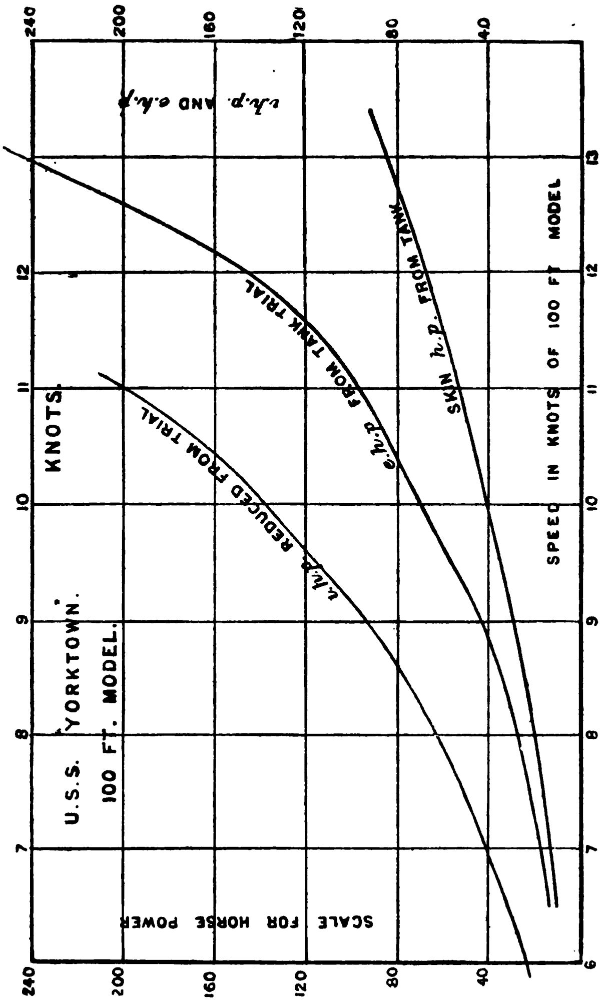
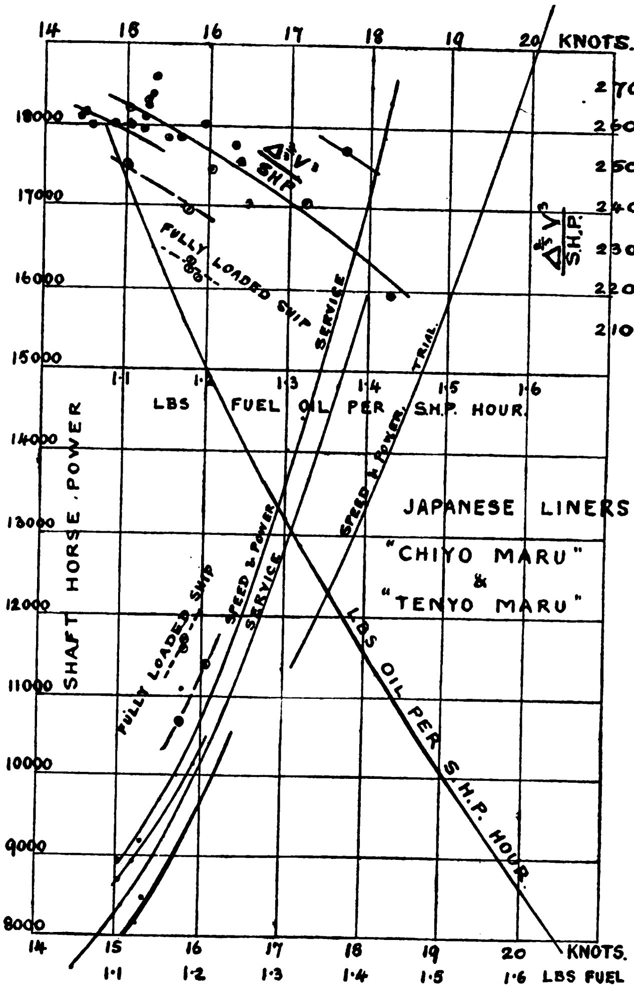


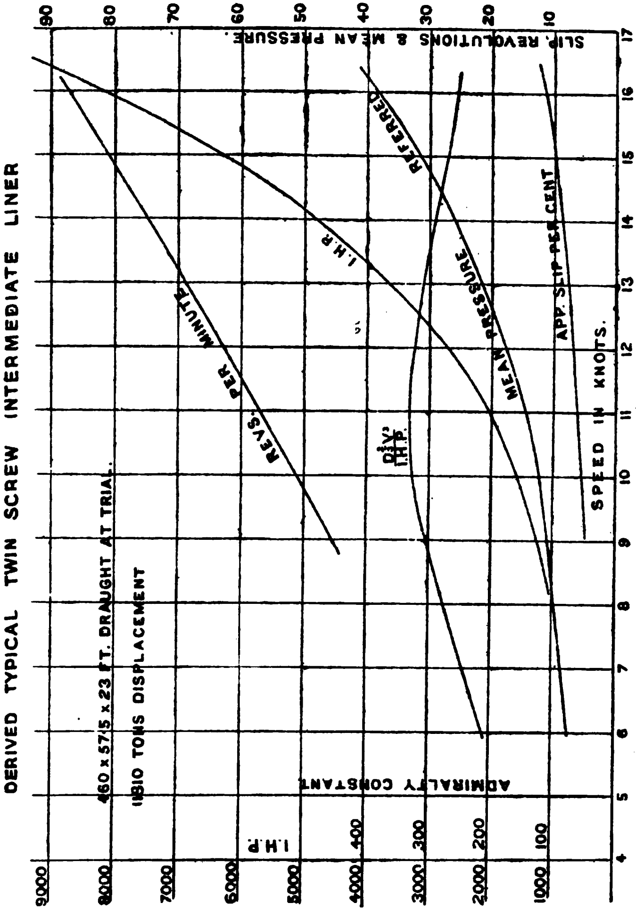
PLATE 23.


U.S.S. "YORKTOWN."
100 FT. MODEL.
KNOTS.
i.h.p. AND e.h.p.
SCALE FOR HORSE POWER
SPEED IN KNOTS OF 100 FT MODEL
i.h.p. REDUCED FROM TRIAL
e.h.p FROM TANK TRIAL
SKIN h.p. FROM TANK
240
200
160
120
80
40
6
7
8
9
10
11
12
13



JAPANESE LINERS
"CHIYO MARU" & "TENYO MARU"
SHAFT HORSE POWER
KNOTS.
LBS FUEL OIL PER S.H.P. HOUR.
$\frac{\Delta^{\frac{2}{3}} V^3}{S.H.P.}$
FULLY LOADED SHIP
SERVICE
SPEED & POWER TRIAL.
SPEED & POWER SERVICE
LBS OIL PER S.H.P. HOUR.
LBS FUEL

PLATE 25.



DERIVED TYPICAL TWIN SCREW INTERMEDIATE LINER
460 x 57.5 x 23 FT. DRAUGHT AT TRIAL.
11810 TONS DISPLACEMENT
SLIP, REVOLUTIONS & MEAN PRESSURE.
ADMIRALTY CONSTANT
I.H.P
SPEED IN KNOTS.
REVS. PER MINUTE
I.H.P
MEAN PRESSURE REFERRED
APP. SLIP PER CENT

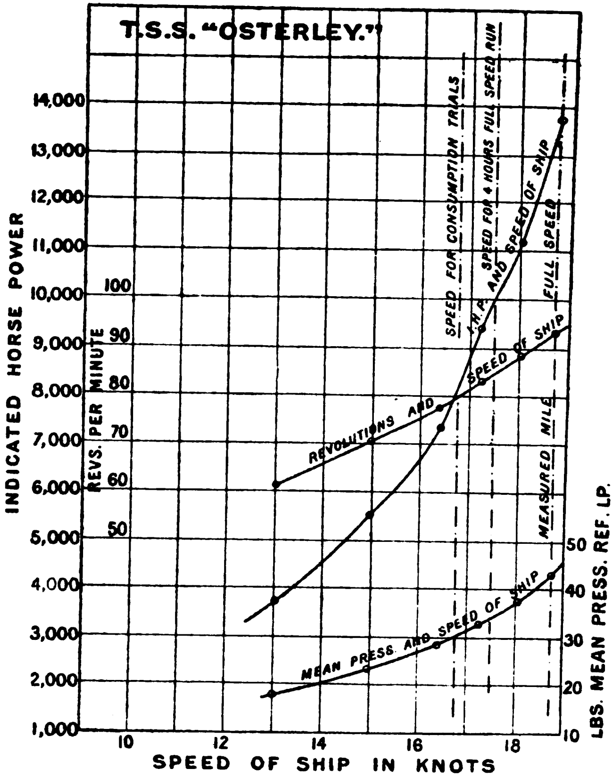
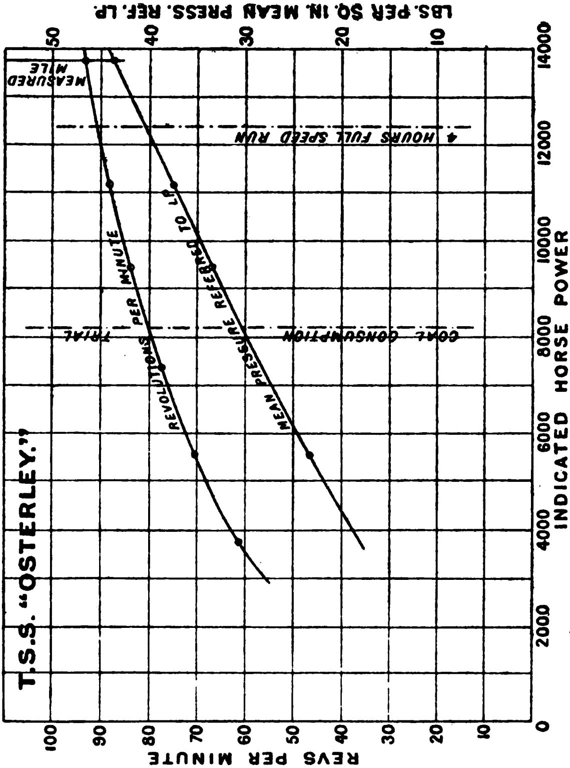

T.S.S. "OSTERLEY."
INDICATED HORSE POWER
14,000
13,000
12,000
11,000
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000
REVS. PER MINUTE
100
90
80
70
60
50
SPEED FOR CONSUMPTION TRIALS
SPEED FOR 4 HOURS FULL SPEED RUN
I.H.P. AND SPEED OF SHIP
FULL SPEED MEASURED MILE
REVOLUTIONS AND SPEED OF SHIP
MEAN PRESS. AND SPEED OF SHIP
LBS. MEAN PRESS. REF. LP.
50
40
30
20
10
10
12
14
16
18
SPEED OF SHIP IN KNOTS

PLATE 27.


T.S.S. "OSTERLEY."
LBS. PER SQ. IN. MEAN PRESS. REF. LP.
50 40 30 20 10
REVS PER MINUTE
100 90 80 70 60 50 40 30 20 10 0
INDICATED HORSE POWER
2000 4000 6000 8000 10000 12000 14000
REVOLUTIONS PER MINUTE
MEAN PRESSURE REFERRED TO LP
MEASURED MILE
4 HOURS FULL SPEED RUN
COAL CONSUMPTION TRIAL

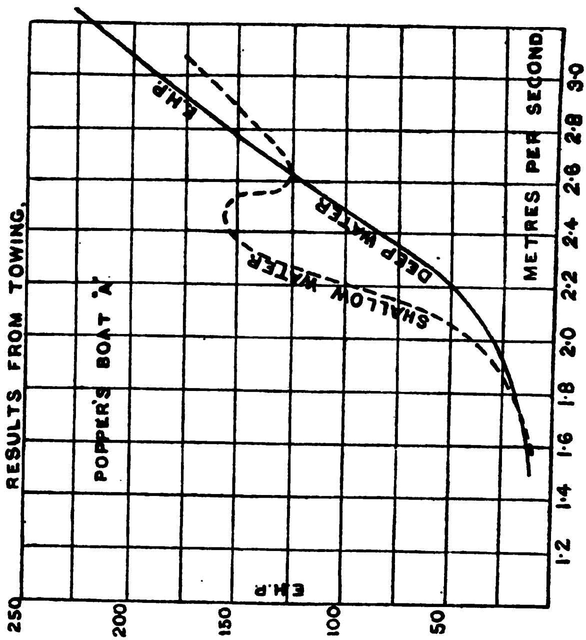

RESULTS FROM TOWING.
POPPER'S BOAT 'A'
E.H.P.
DEEP WATER
SHALLOW WATER
METRES PER SECOND
1·2 1·4 1·6 1·8 2·0 2·2 2·4 2·6 2·8 3·0
E.H.P
250 200 150 100 50

PLATE 29.

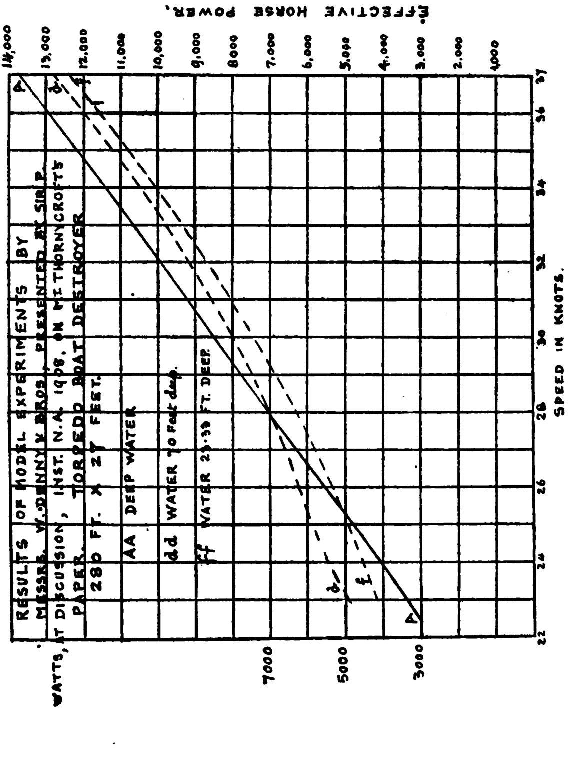
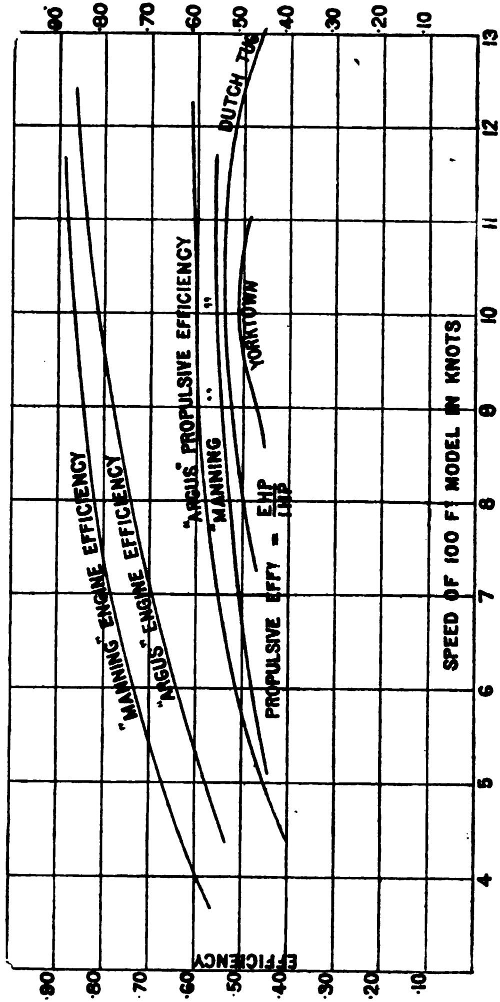

RESULTS OF MODEL EXPERIMENTS BY MESSRS. W. DENNY & BROS. PRESENTED BY SIR P. WATTS, AT DISCUSSION, INST. N.A. 1906, ON MR THORNYCROFT'S PAPER. TORPEDO BOAT DESTROYER
280 FT. X 27 FEET.
AA DEEP WATER
dd WATER 70 Feet deep.
ff WATER 25·39 FT. DEEP
EFFECTIVE HORSE POWER.
SPEED IN KNOTS.

PLATE 30.


"MANNING" ENGINE EFFICIENCY
"ARGUS" ENGINE EFFICIENCY
"ARGUS" PROPULSIVE EFFICIENCY
"MANNING" " "
DUTCH TUG
YORKTOWN
PROPULSIVE EFFY = EHP / IHP
SPEED OF 100 FT MODEL IN KNOTS
EFFICIENCY
4
5
6
7
8
9
10
11
12
13
·90
·80
·70
·60
·50
·40
·30
·20
·10

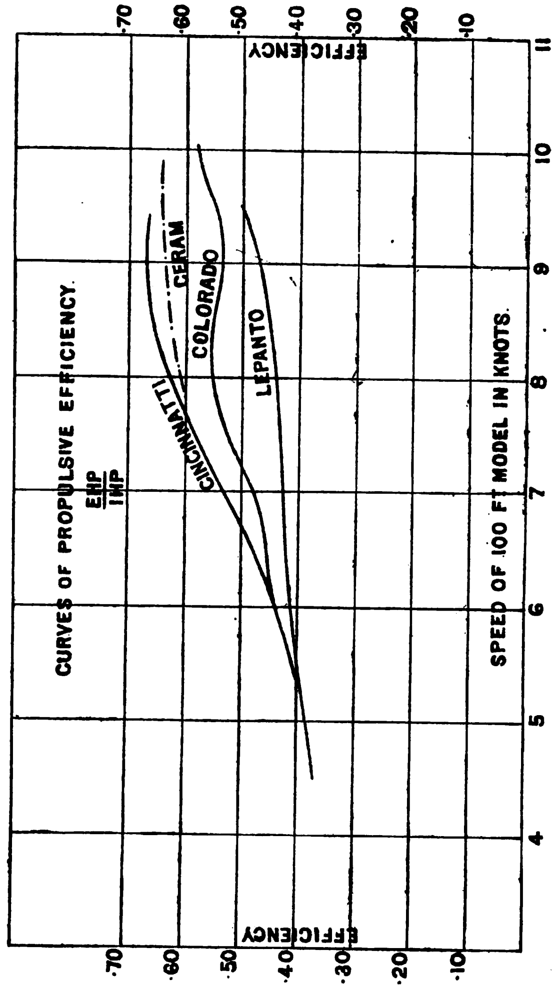
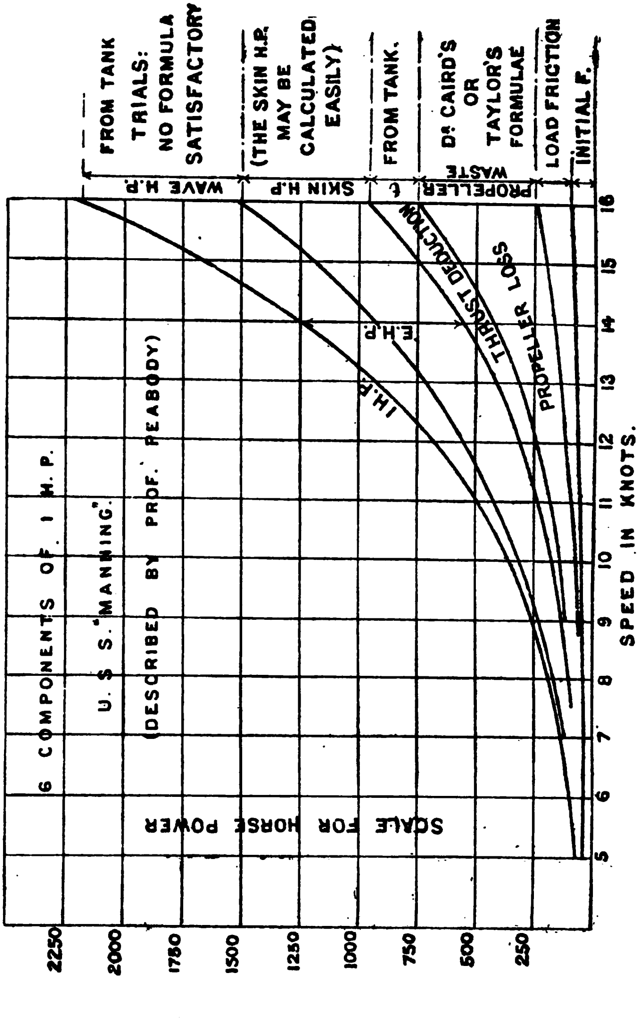

CURVES OF PROPULSIVE EFFICIENCY.
EHP / IHP
SPEED OF 100 FT MODEL IN KNOTS.
EFFICIENCY
EFFICIENCY
CINCINNATTI
CERAM
COLORADO
LEPANTO
·70
·60
·50
·40
·30
·20
·10
4
5
6
7
8
9
10
11

PLATE 32.


6 COMPONENTS OF I. H. P.
U. S. S. "MANNING"
(DESCRIBED BY PROF. PEABODY)
SCALE FOR HORSE POWER
SPEED IN KNOTS.
WAVE H.P.
FROM TANK TRIALS: NO FORMULA SATISFACTORY
SKIN H.P
(THE SKIN H.P. MAY BE CALCULATED EASILY)
t
FROM TANK.
PROPELLER WASTE
Dr. CAIRD'S OR TAYLOR'S FORMULAE
LOAD FRICTION
INITIAL F.
I.H.P.
E.H.P.
THRUST DEDUCTION
PROPELLER LOSS
2250
2000
1750
1500
1250
1000
750
500
250
5
6
7
8
9
10
11
12
13
14
15
16

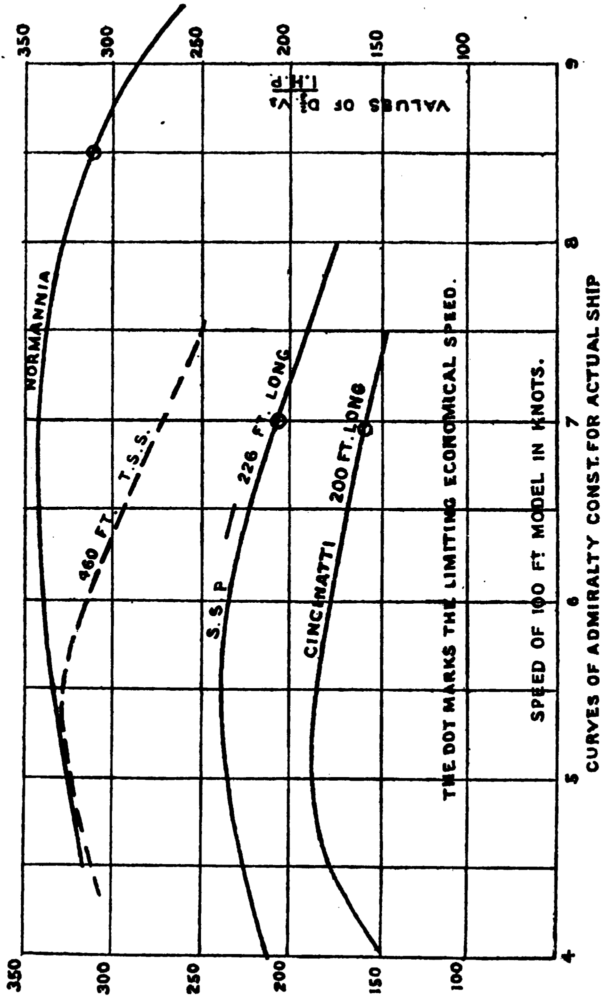
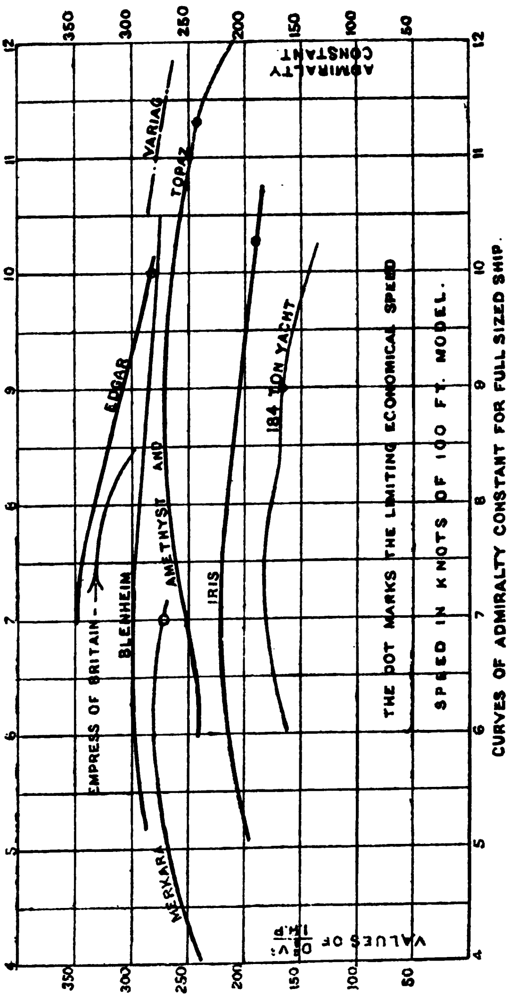
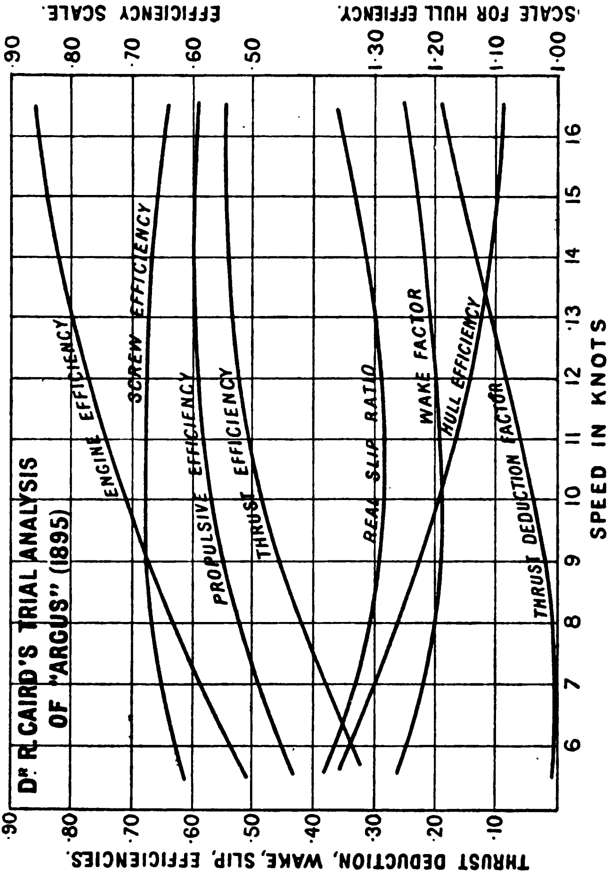
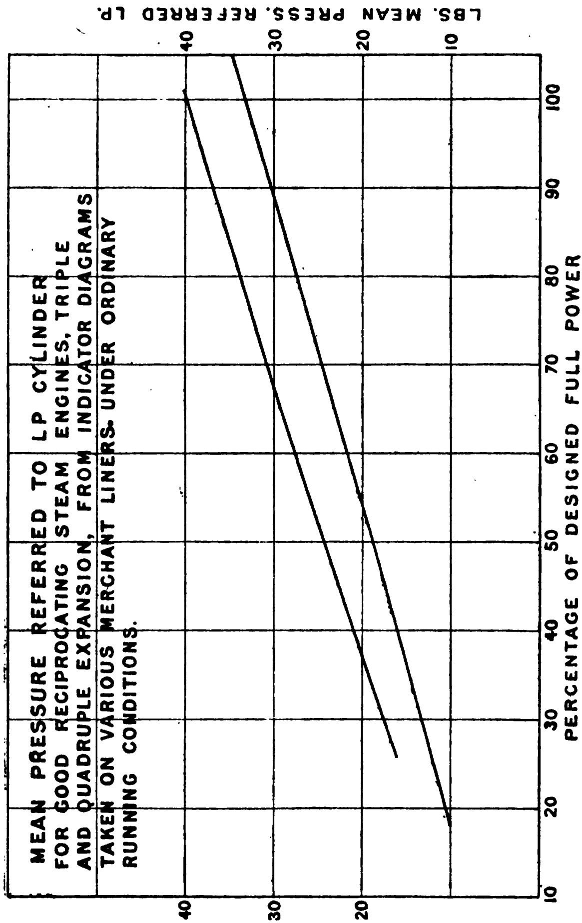

CURVES OF ADMIRALTY CONST. FOR ACTUAL SHIP
SPEED OF 100 FT. MODEL IN KNOTS.
THE DOT MARKS THE LIMITING ECONOMICAL SPEED.
VALUES OF $\frac{D^{\frac{2}{3}} V^3}{I.H.P.}$
NORMANNIA
460 FT. T.S.S.
S.S.P. 226 FT. LONG
CINCINATTI 200 FT. LONG
350
300
250
200
150
100
4
5
6
7
8
9

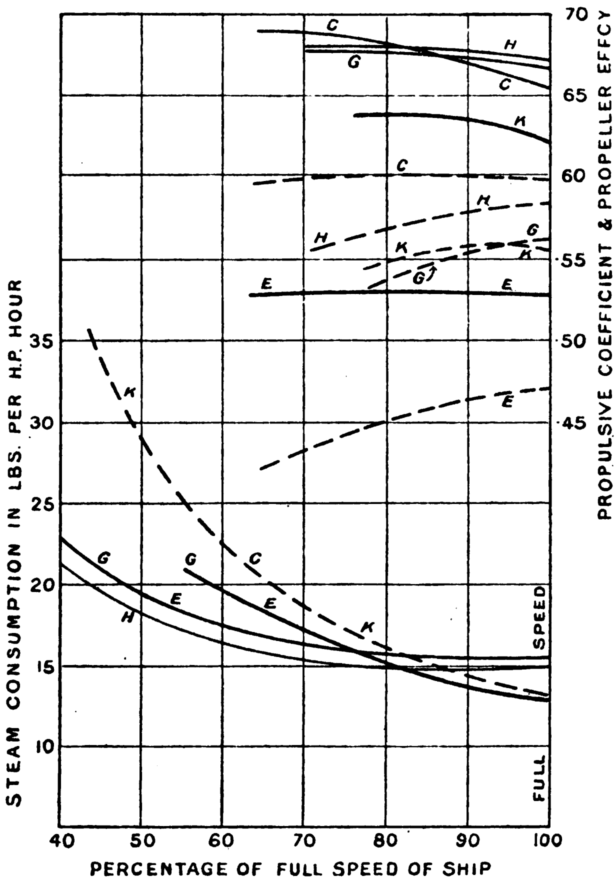
PLATE 34.

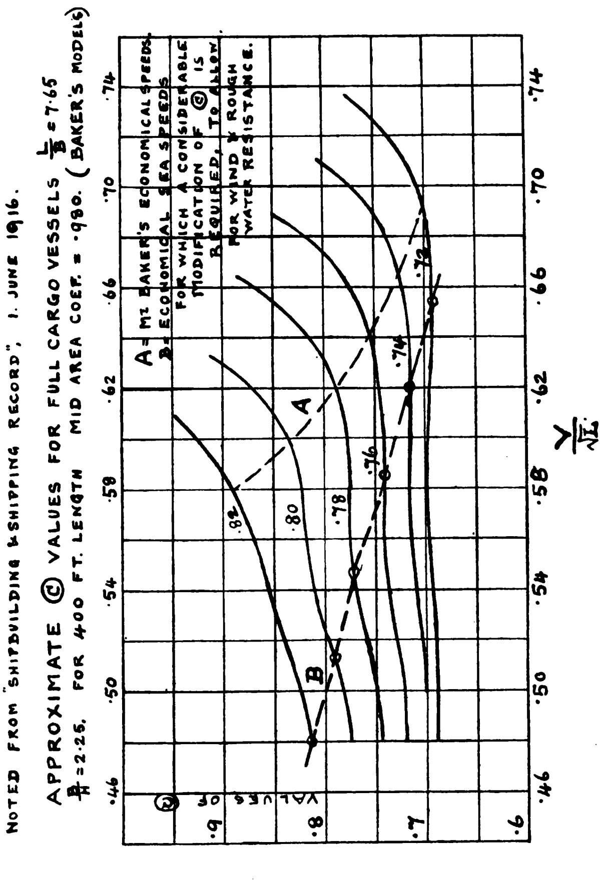


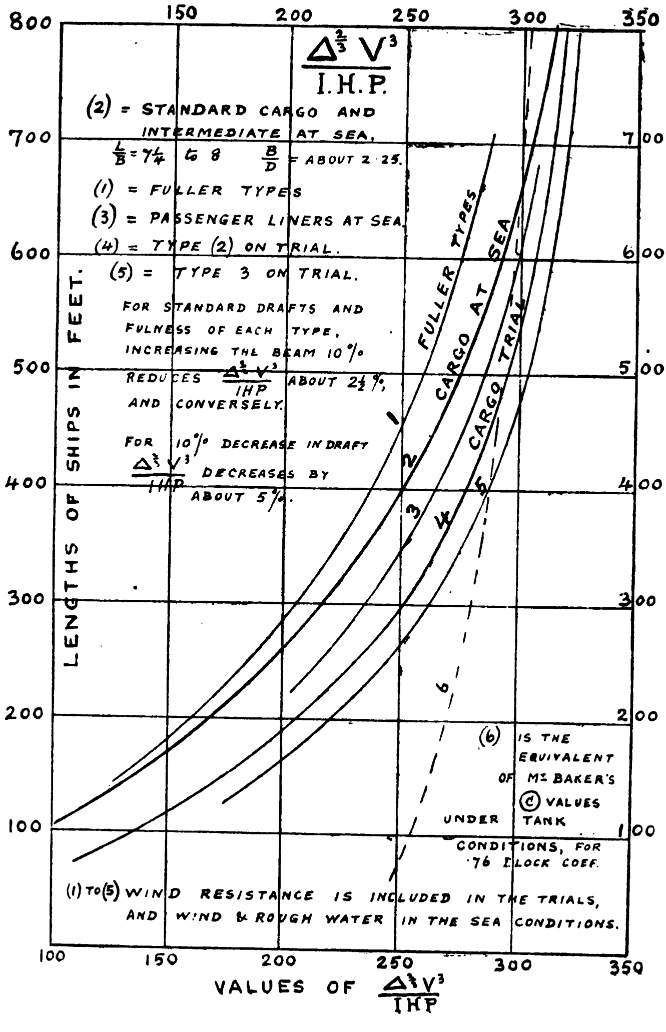
ADMIRALTY CONSTANT
VARIAG
TOPAZ
EDGAR
184 TON YACHT
AMETHYST AND
EMPRESS OF BRITAIN
BLENHEIM
IRIS
MERKARA
VALUES OF $\frac{D^{\frac{2}{3}}V^3}{I.H.P.}$
THE DOT MARKS THE LIMITING ECONOMICAL SPEED
SPEED IN KNOTS OF 100 FT. MODEL.
CURVES OF ADMIRALTY CONSTANT FOR FULL SIZED SHIP.


Dr. R. CAIRD'S TRIAL ANALYSIS OF "ARGUS" (1895)
EFFICIENCY SCALE.
SCALE FOR HULL EFFIENCY.
THRUST DEDUCTION, WAKE, SLIP, EFFICIENCIES.
SPEED IN KNOTS
ENGINE EFFICIENCY
SCREW EFFICIENCY
PROPULSIVE EFFICIENCY
THRUST EFFICIENCY
REAL SLIP RATIO
WAKE FACTOR
HULL EFFICIENCY
THRUST DEDUCTION FACTOR
·90
·80
·70
·60
·50
·40
·30
·20
·10
1·30
1·20
1·10
1·00
6
7
8
9
10
11
12
13
14
15
16

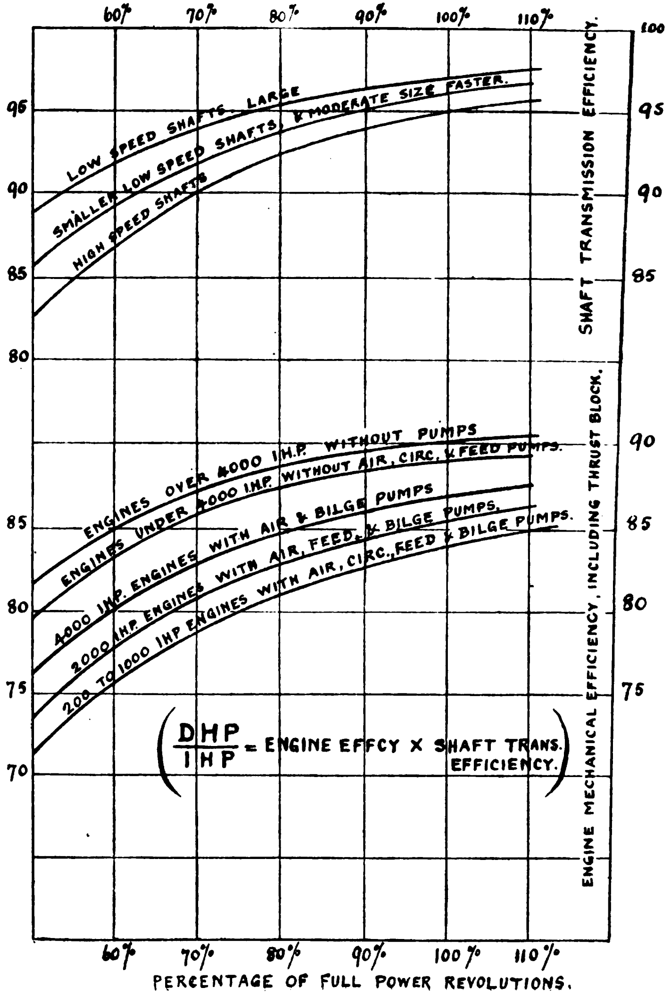

MEAN PRESSURE REFERRED TO LP CYLINDER FOR GOOD RECIPROCATING STEAM ENGINES, TRIPLE AND QUADRUPLE EXPANSION, FROM INDICATOR DIAGRAMS TAKEN ON VARIOUS MERCHANT LINERS. UNDER ORDINARY RUNNING CONDITIONS.
LBS. MEAN PRESS. REFERRED LP.
PERCENTAGE OF DESIGNED FULL POWER
10
20
30
40
50
60
70
80
90
100
10
20
30
40

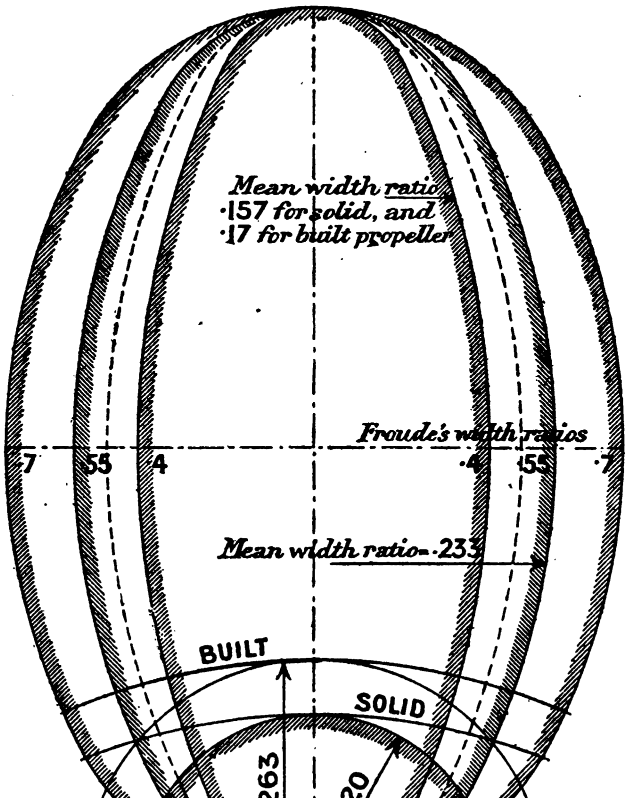

STEAM CONSUMPTION IN LBS. PER H.P. HOUR
PROPULSIVE COEFFICIENT & PROPELLER EFFCY
PERCENTAGE OF FULL SPEED OF SHIP
FULL SPEED
40
50
60
70
80
90
100
10
15
20
25
30
35
·45
·50
·55
60
65
70
C
H
G
K
E

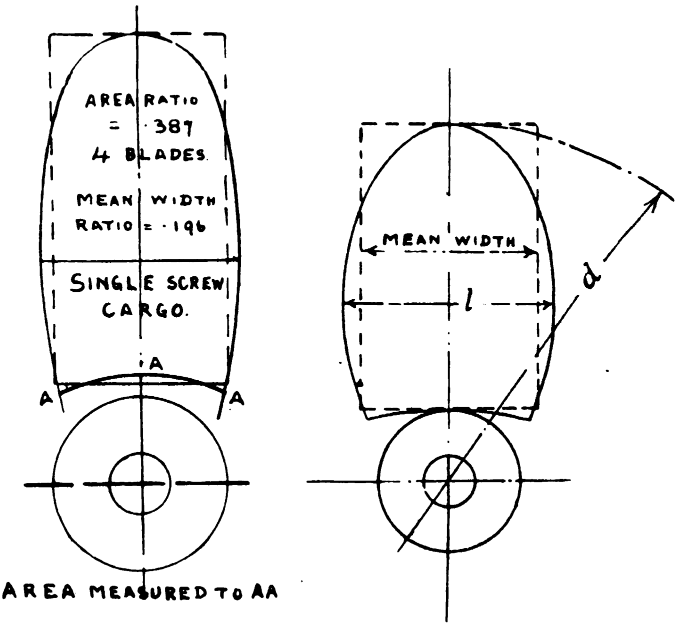

NOTED FROM "SHIPBUILDING & SHIPPING RECORD", 1. JUNE 1916.
APPROXIMATE (C) VALUES FOR FULL CARGO VESSELS L/B = 7·65
B/H = 2·25. FOR 400 FT. LENGTH MID AREA COEF. = ·980. (BAKER'S MODELS)
A = Mr BAKER'S ECONOMICAL SPEEDS.
B = ECONOMICAL SEA SPEEDS FOR WHICH A CONSIDERABLE MODIFICATION OF (C) IS REQUIRED, TO ALLOW FOR WIND & ROUGH WATER RESISTANCE.
A
B
·82
·80
·78
·76
·74
·72
·46
·50
·54
·58
·62
·66
·70
·74
V/√L
VALUES OF (C)
·9
·8
·7
·6

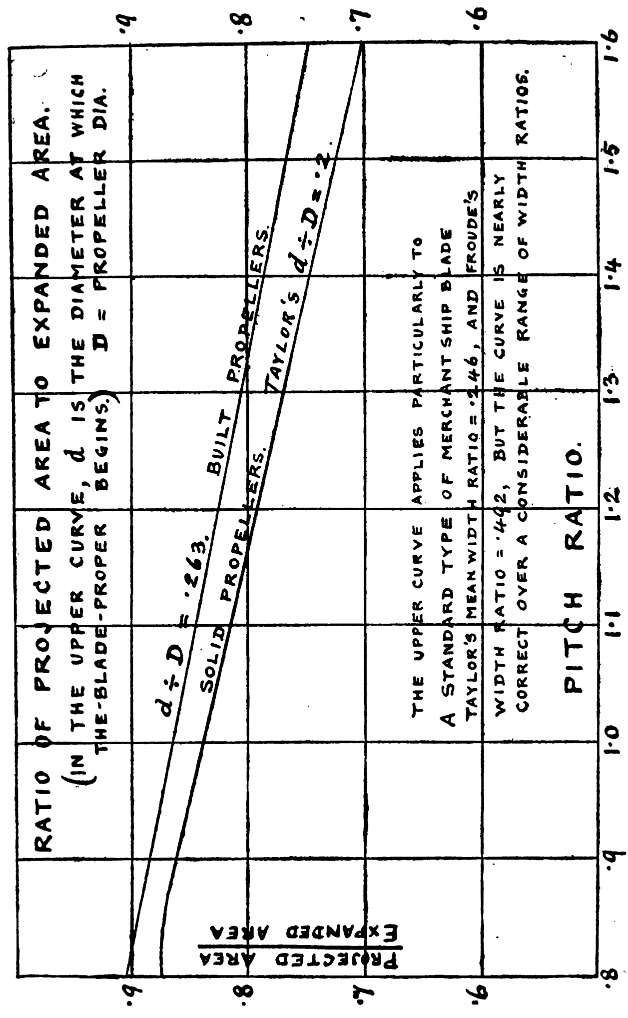


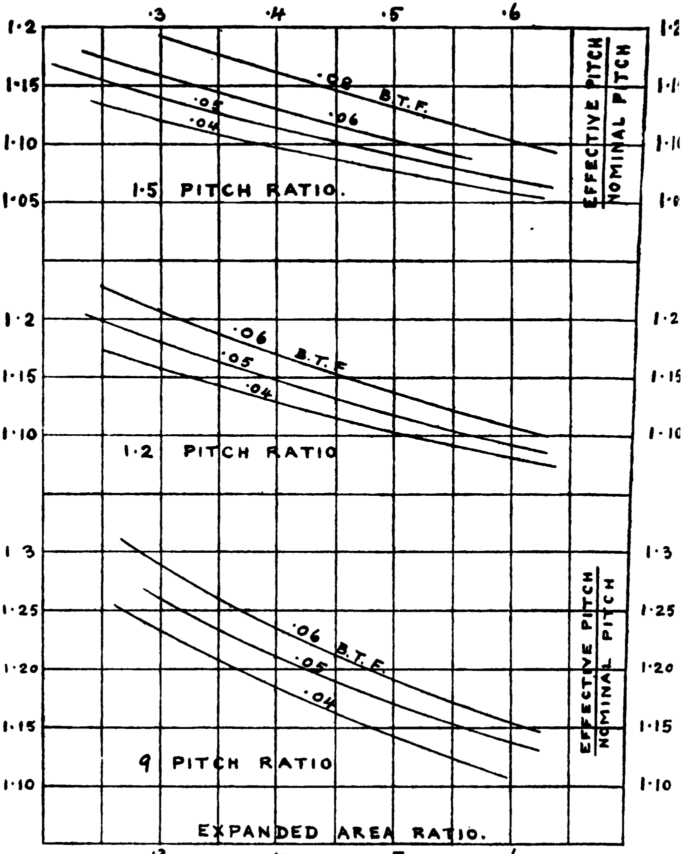
Δ⅔V³ / I.H.P.
(2) = STANDARD CARGO AND INTERMEDIATE AT SEA,
L/B = 7¼ to 8   B/D = ABOUT 2·25.
(1) = FULLER TYPES
(3) = PASSENGER LINERS AT SEA
(4) = TYPE (2) ON TRIAL.
(5) = TYPE 3 ON TRIAL.
FOR STANDARD DRAFTS AND FULNESS OF EACH TYPE, INCREASING THE BEAM 10% REDUCES Δ⅔V³/IHP ABOUT 2½%, AND CONVERSELY.
FOR 10% DECREASE IN DRAFT Δ⅔V³/IHP DECREASES BY ABOUT 5%.
FULLER TYPES
CARGO AT SEA
CARGO TRIAL
1
2
3
4
5
6
(6) IS THE EQUIVALENT OF Mr BAKER'S Ⓒ VALUES UNDER TANK CONDITIONS, FOR ·76 BLOCK COEF.
(1) TO (5) WIND RESISTANCE IS INCLUDED IN THE TRIALS, AND WIND & ROUGH WATER IN THE SEA CONDITIONS.
LENGTHS OF SHIPS IN FEET.
VALUES OF Δ⅔V³/IHP
100
150
200
250
300
350
800
700
600
500
400
300
200
100


60% 70% 80% 90% 100% 110%
100
95
90
85
80
75
70
LOW SPEED SHAFTS. LARGE
SMALLER LOW SPEED SHAFTS, & MODERATE SIZE FASTER.
HIGH SPEED SHAFTS
SHAFT TRANSMISSION EFFICIENCY.
ENGINES OVER 4000 I.H.P. WITHOUT PUMPS
ENGINES UNDER 4000 I.H.P. WITHOUT AIR, CIRC. & FEED PUMPS.
4000 I.H.P. ENGINES WITH AIR & BILGE PUMPS
2000 I.H.P. ENGINES WITH AIR, FEED, & BILGE PUMPS.
200 TO 1000 I.H.P. ENGINES WITH AIR, CIRC., FEED & BILGE PUMPS.
ENGINE MECHANICAL EFFICIENCY, INCLUDING THRUST BLOCK.
(DHP / IHP = ENGINE EFFCY X SHAFT TRANS. EFFICIENCY.)
PERCENTAGE OF FULL POWER REVOLUTIONS.


Mean width ratio
·157 for solid, and
·17 for built propeller
Froude's width ratios
·7
·55
·4
·4
·55
·7
Mean width ratio = ·233
BUILT
SOLID
263
20

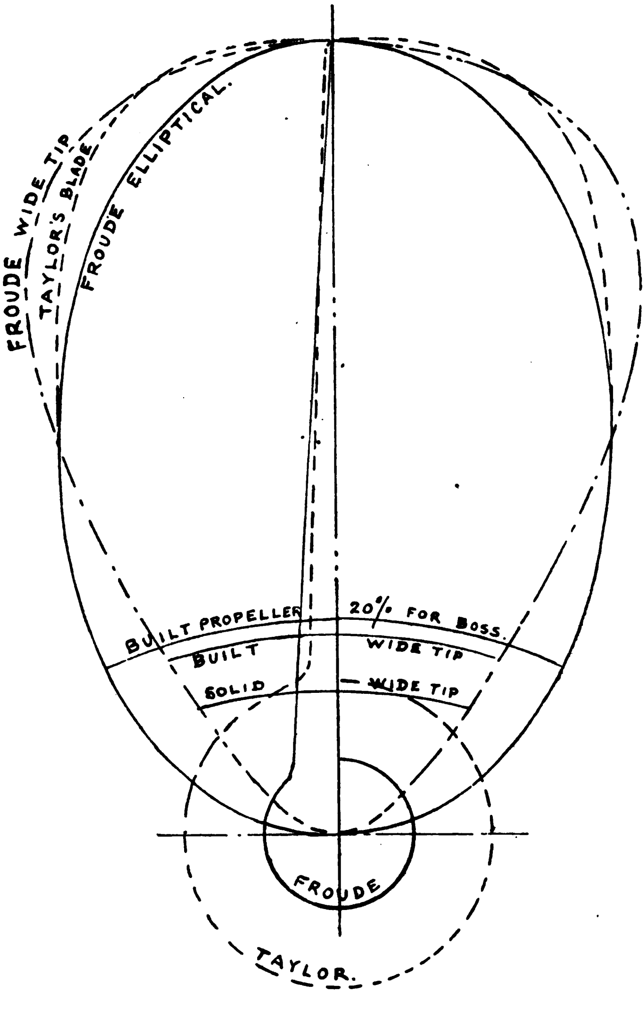

AREA RATIO
= ·387
4 BLADES.
MEAN WIDTH
RATIO = ·196
SINGLE SCREW
CARGO.
A
A
A
AREA MEASURED TO AA
MEAN WIDTH
l
d


RATIO OF PROJECTED AREA TO EXPANDED AREA.
(IN THE UPPER CURVE, d IS THE DIAMETER AT WHICH THE-BLADE-PROPER BEGINS.) D = PROPELLER DIA.
d ÷ D = ·263.
BUILT PROPELLERS.
SOLID PROPELLERS.
TAYLOR'S d ÷ D = ·2
THE UPPER CURVE APPLIES PARTICULARLY TO
A STANDARD TYPE OF MERCHANT SHIP BLADE
TAYLOR'S MEAN WIDTH RATIO = ·246, AND FROUDE'S
WIDTH RATIO = ·402, BUT THE CURVE IS NEARLY
CORRECT OVER A CONSIDERABLE RANGE OF WIDTH RATIOS.
PITCH RATIO.
PROJECTED AREA / EXPANDED AREA
·8
·9
1·0
1·1
1·2
1·3
1·4
1·5
1·6
·6
·7
·8
·9



3 BLADED PROPELLERS, ELLIPTICAL BLADES.

BOSS DIA = ·2 × PROPELLER DIA


FROUDE WIDE TIP
TAYLOR'S BLADE
FROUDE ELLIPTICAL.
BUILT PROPELLER
20% FOR BOSS.
BUILT
WIDE TIP
SOLID
WIDE TIP
FROUDE
TAYLOR.

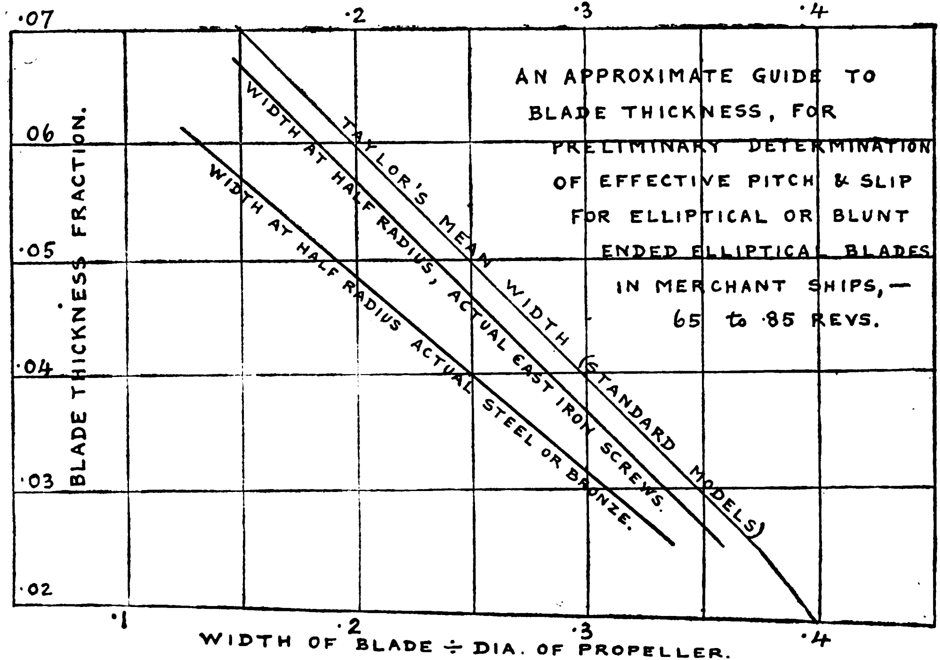
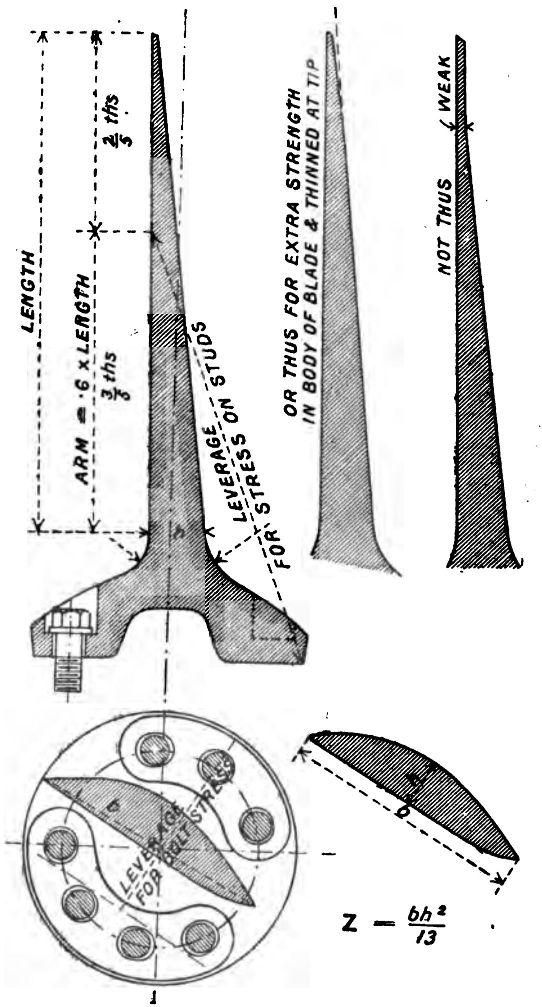
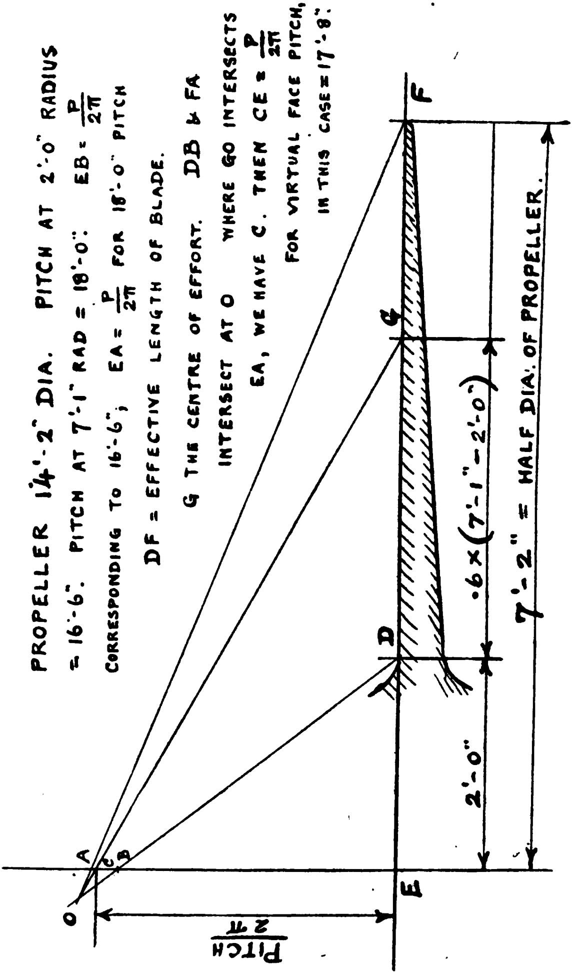
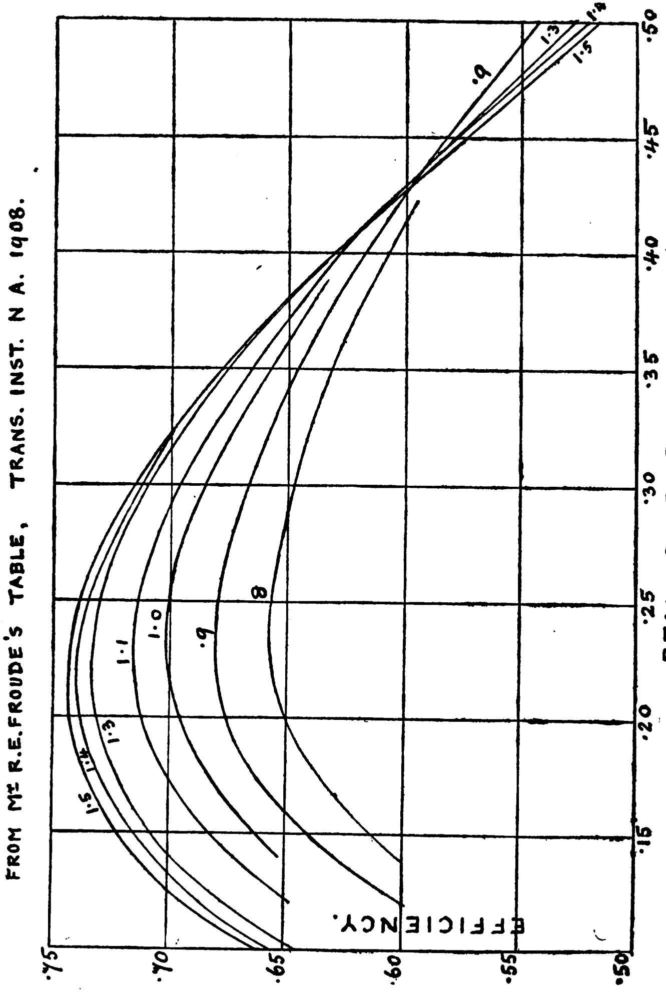

AN APPROXIMATE GUIDE TO
BLADE THICKNESS, FOR
PRELIMINARY DETERMINATION
OF EFFECTIVE PITCH & SLIP
FOR ELLIPTICAL OR BLUNT
ENDED ELLIPTICAL BLADES
IN MERCHANT SHIPS,—
65 to ·85 REVS.
TAYLOR'S MEAN WIDTH (STANDARD MODELS)
WIDTH AT HALF RADIUS, ACTUAL CAST IRON SCREWS.
WIDTH AT HALF RADIUS ACTUAL STEEL OR BRONZE.
BLADE THICKNESS FRACTION.
·07
·06
·05
·04
·03
·02
·1
·2
·3
·4
WIDTH OF BLADE ÷ DIA. OF PROPELLER.

PLATE 47.



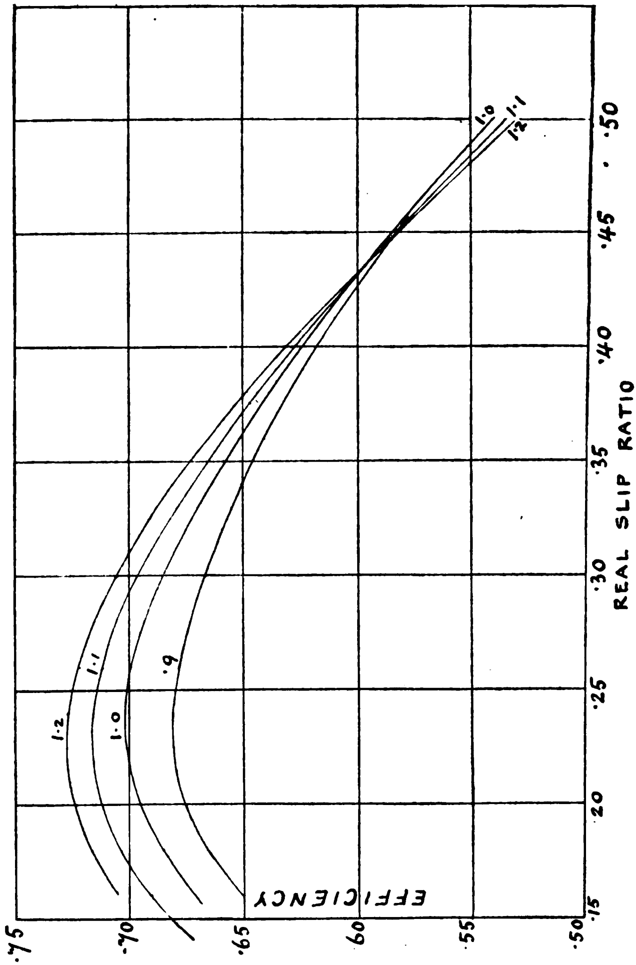
LENGTH
ARM = ·6 x LENGTH
$\frac{2}{5}$ ths
$\frac{3}{5}$ ths
LEVERAGE FOR STRESS ON STUDS
OR THUS FOR EXTRA STRENGTH IN BODY OF BLADE & THINNED AT TIP
NOT THUS
WEAK
LEVERAGE FOR BOLT STRESS
h
b
$Z = \frac{bh^2}{13}$


PROPELLER 14'-2" DIA. PITCH AT 2'-0" RADIUS
= 16'-6". PITCH AT 7'-1" RAD = 18'-0". EB = P/2π
CORRESPONDING TO 16'-6"; EA = P/2π FOR 18'-0" PITCH
DF = EFFECTIVE LENGTH OF BLADE.
G THE CENTRE OF EFFORT. DB & FA
INTERSECT AT O WHERE GO INTERSECTS
EA, WE HAVE C. THEN CE = P/2π
FOR VIRTUAL FACE PITCH,
IN THIS CASE = 17'-8".
F
G
D
E
A
C
B
O
PITCH/2π
2'-0"
·6×(7'-1"-2'-0")
7'-2" = HALF DIA. OF PROPELLER.

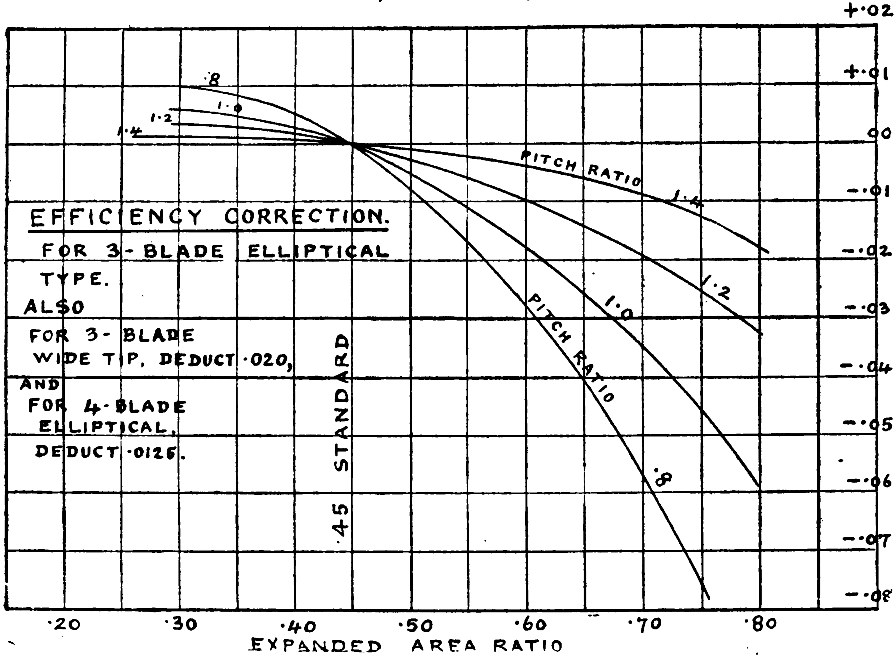


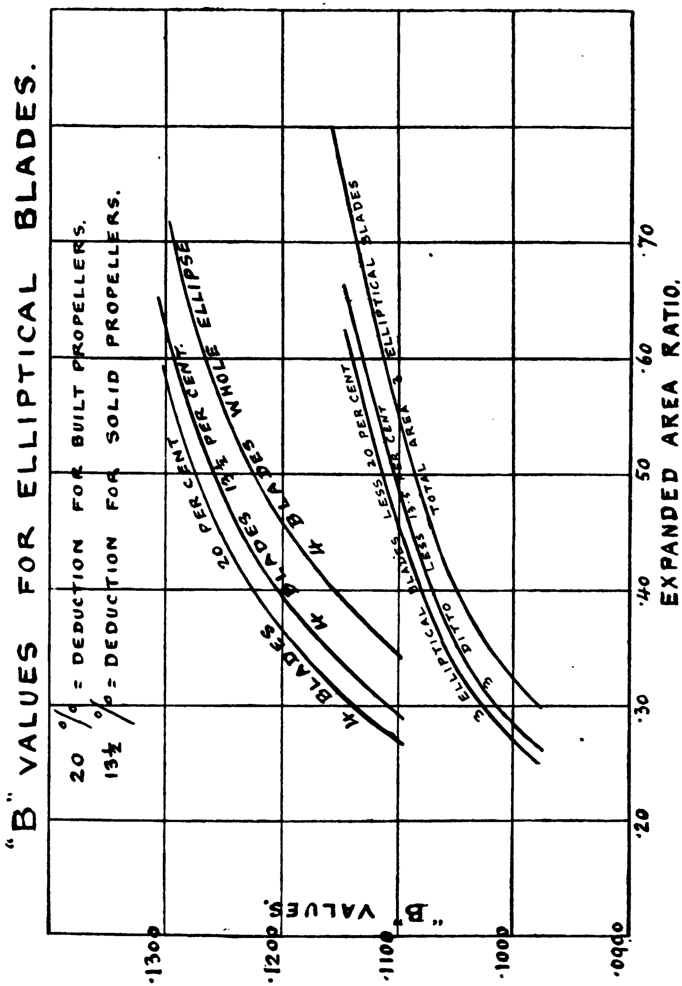
Fyfe, Steamship Coefficients.—E. & F. N. SPON, LTD.

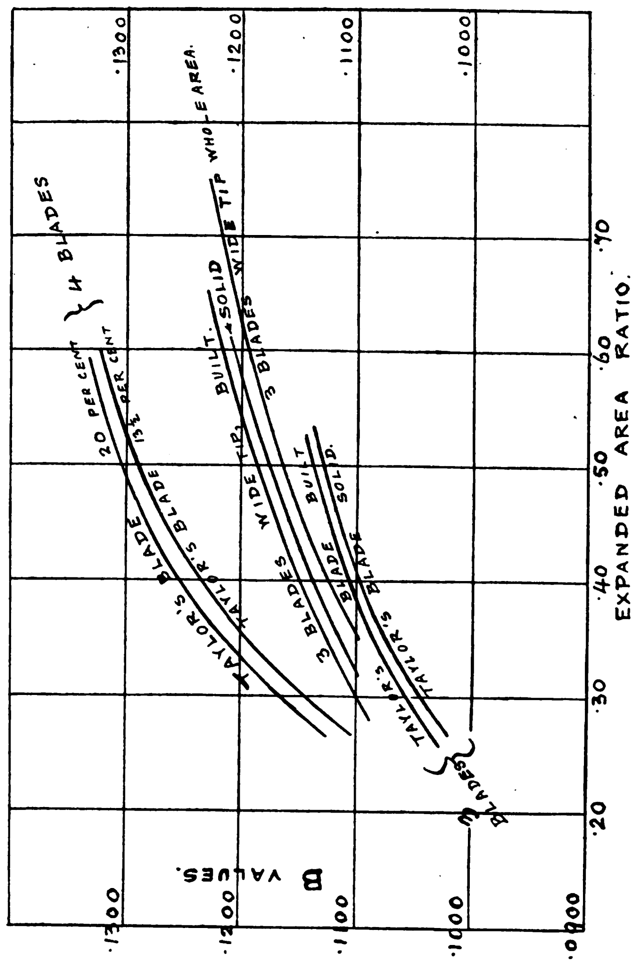
PLOTTED FROM Mr R.E. FROUDE'S TABLE, TRANS. I.N.A. 1908.



FROM MR R. E. FROUDE'S 1908 PAPER, INST. N.A.



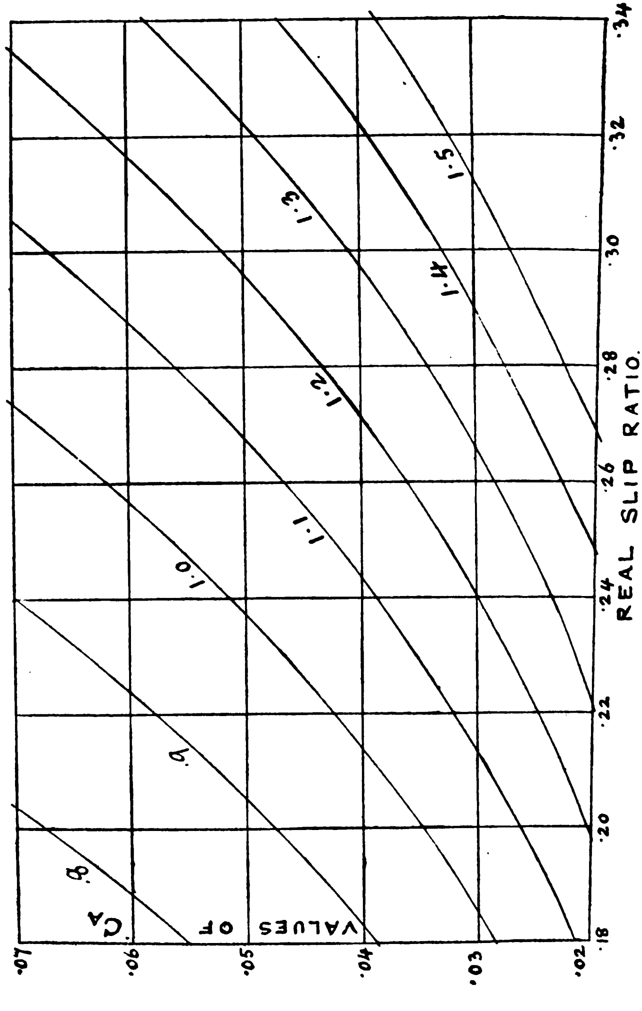

"B" VALUES FOR ELLIPTICAL BLADES.
20 % = DEDUCTION FOR BUILT PROPELLERS.
13½ % = DEDUCTION FOR SOLID PROPELLERS.
4 BLADES WHOLE ELLIPSE
4 BLADES 13½ PER CENT.
4 BLADES 20 PER CENT
ELLIPTICAL BLADES
3 ELLIPTICAL BLADES LESS 13½ PER CENT TOTAL AREA
3 DITTO LESS 20 PER CENT
·1300
·1200
·1100
·1000
·0900
"B" VALUES.
·20
·30
·40
·50
·60
·70
EXPANDED AREA RATIO.

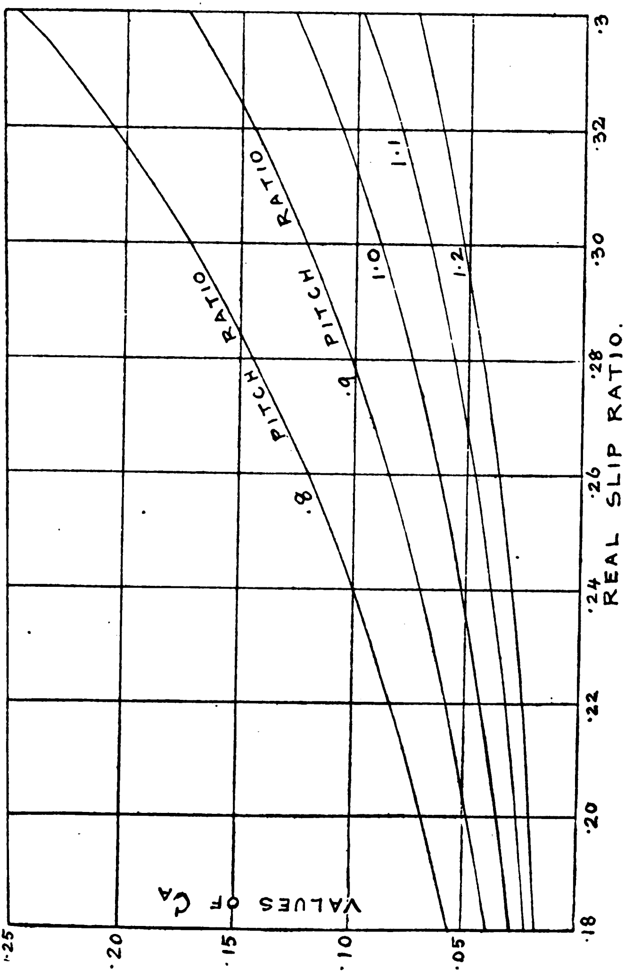


4 BLADES
20 PER CENT TAYLOR'S BLADE
13½ PER CENT TAYLOR'S BLADE
BUILT.
4 SOLID
WIDE TIP WHOLE AREA
3 BLADES
WIDE TIP
3 BLADES
BUILT
SOLID
3 BLADE
TAYLOR'S BLADE
TAYLOR'S
3 BLADES
B VALUES.
·1300
·1200
·1100
·1000
·0900
·20
·30
·40
·50
·60
·70
EXPANDED AREA RATIO.

PLATE 54.

FROM R.E. FROUDE, 1908, TRANS. INST. N.A.




VALUES OF C_A
REAL SLIP RATIO.
PITCH RATIO .8
PITCH RATIO .9
1.0
1.1
1.2
·25
·20
·15
·10
·05
·18
·20
·22
·24
·26
·28
·30
·32
·3

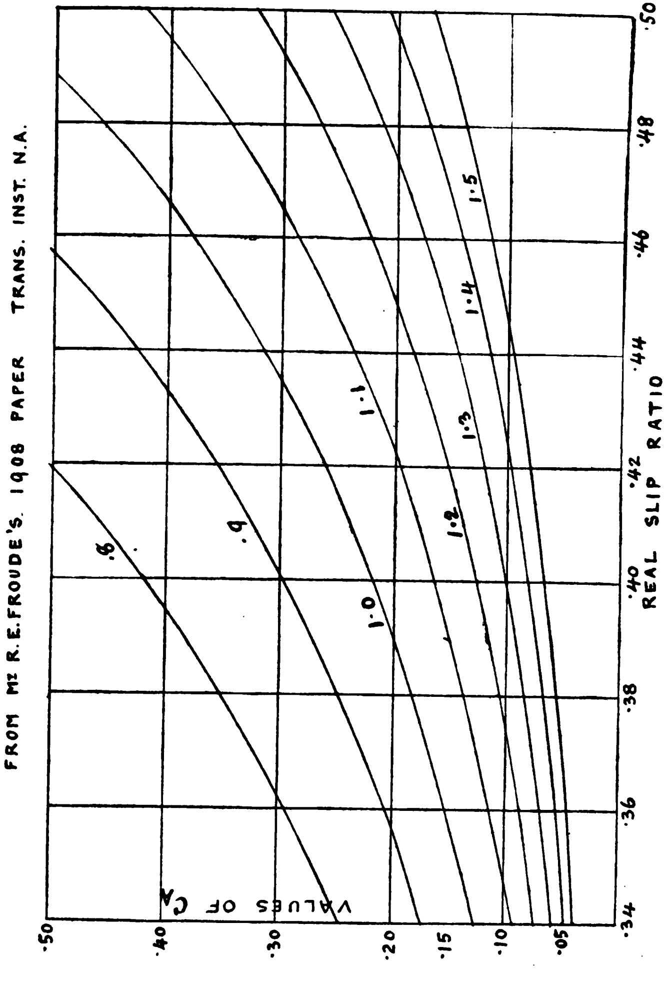
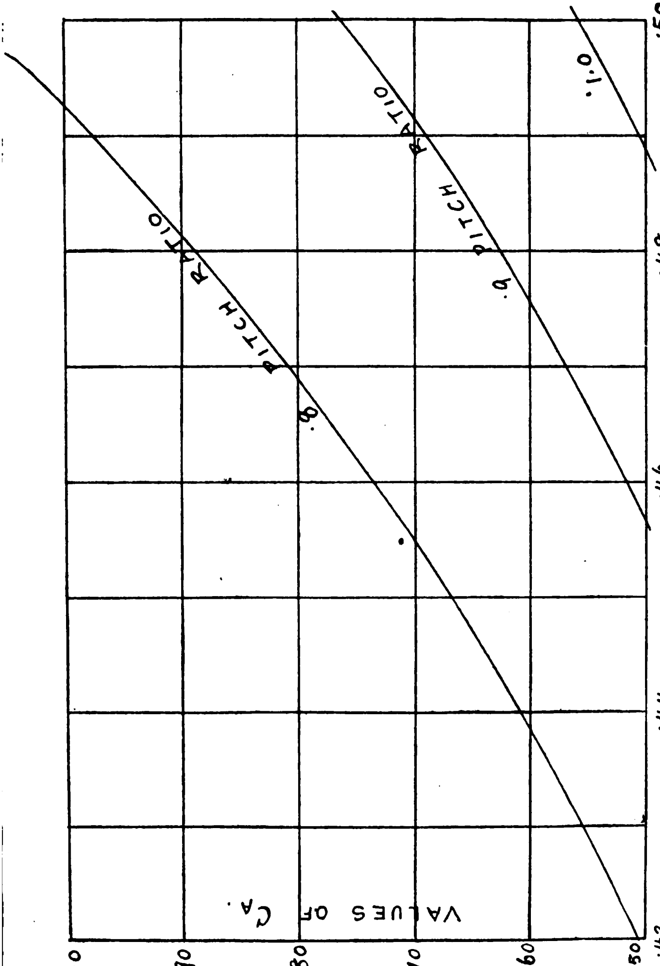
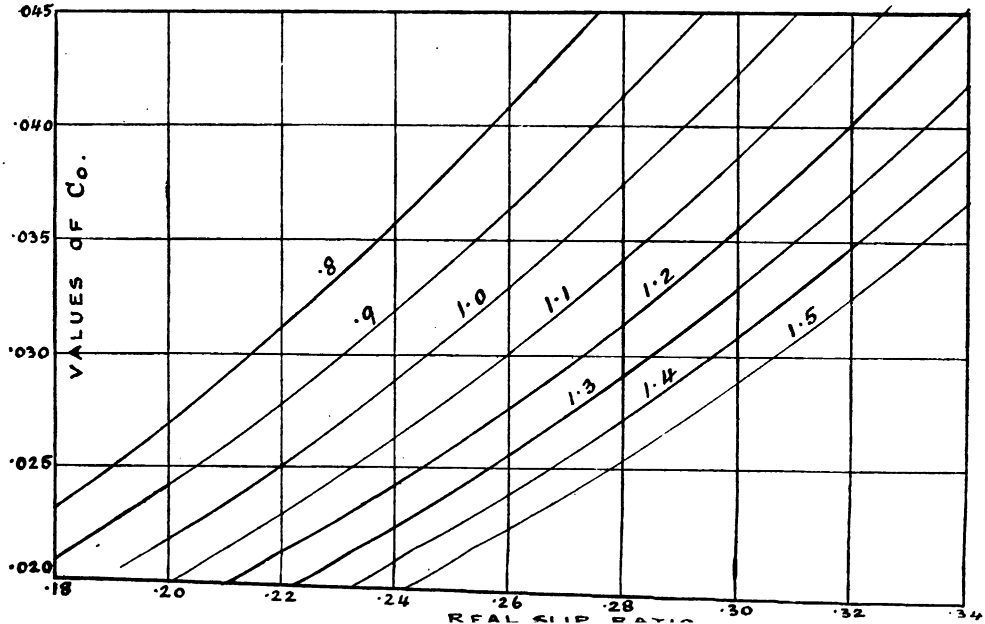
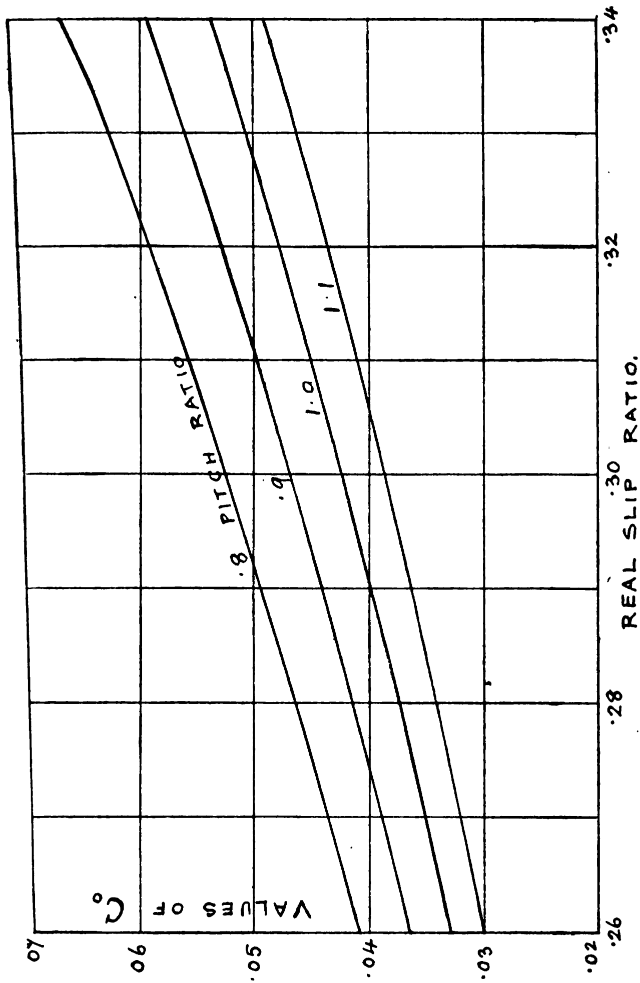

FROM Mr R. E. FROUDE'S 1908 PAPER TRANS. INST. N.A.
REAL SLIP RATIO
VALUES OF CA
·34
·36
·38
·40
·42
·44
·46
·48
·50
·50
·40
·30
·20
·15
·10
·05
·8
·9
1·0
1·1
1·2
1·3
1·4
1·5

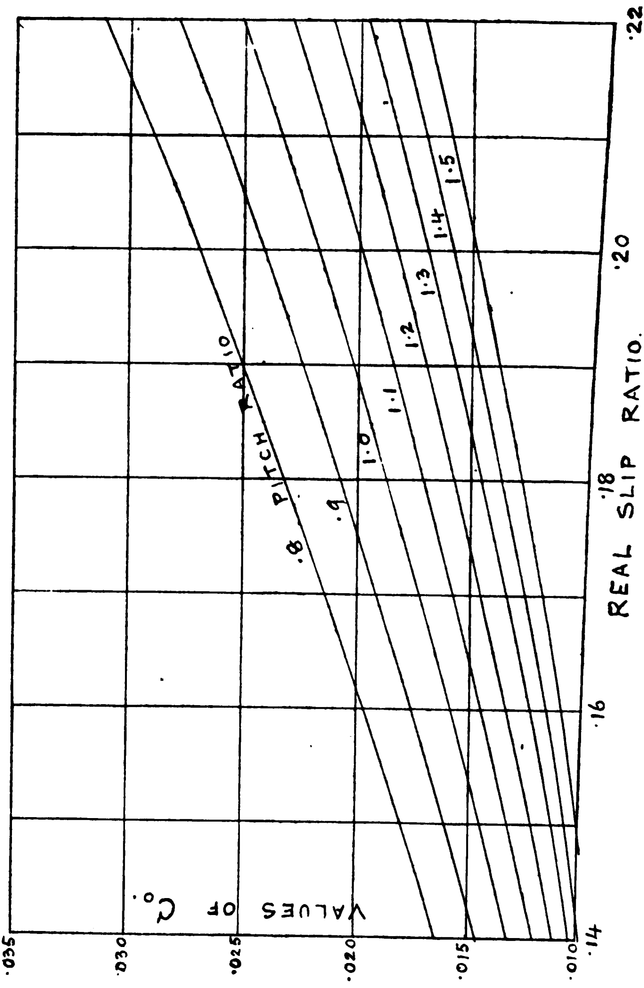
PLATE 57.


REAL SLIP RATIO.
VALUES OF C_A.
·8 PITCH RATIO
·9 PITCH RATIO
1.0
·42
·44
·46
·48
·50
1·0
·90
·80
·70
·60
·50

FROM Mr R.E FROUDE'S 1908 PAPER. I.N.A.




VALUES OF C_0
.07
.06
.05
.04
.03
.02
.26
.28
.30
.32
.34
REAL SLIP RATIO.
.8 PITCH RATIO
.9
1.0
1.1


REAL SLIP RATIO.
VALUES OF C_o.
PITCH RATIO
·8
·9
1·0
1·1
1·2
1·3
1·4
1·5
·14
·16
·18
·20
·22
·035
·030
·025
·020
·015
·010

FROM MR. R. E. FROUDE'S TABLE, 1908 TRANS. INST. N.A.

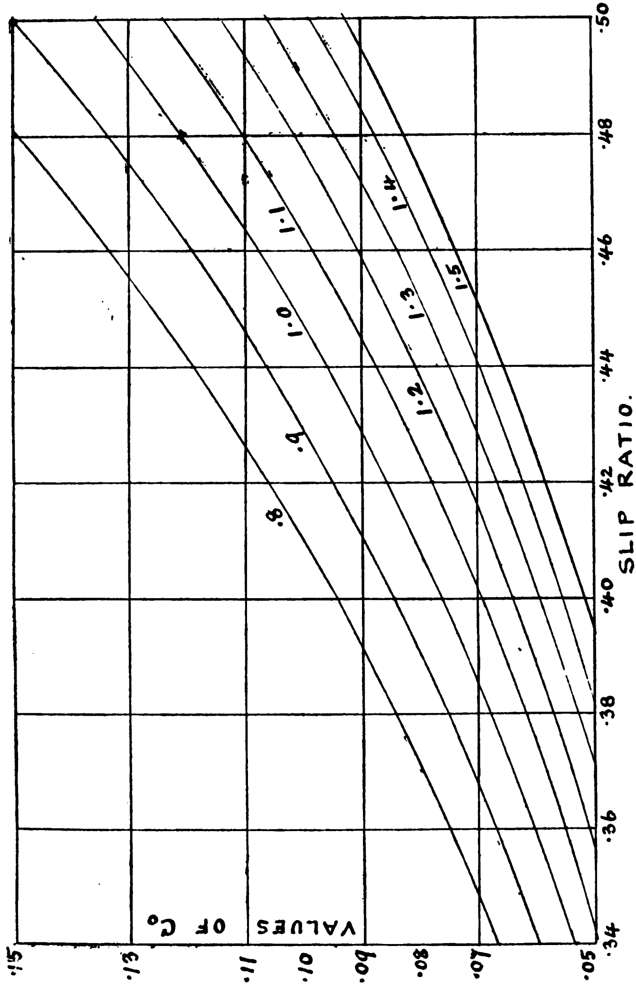
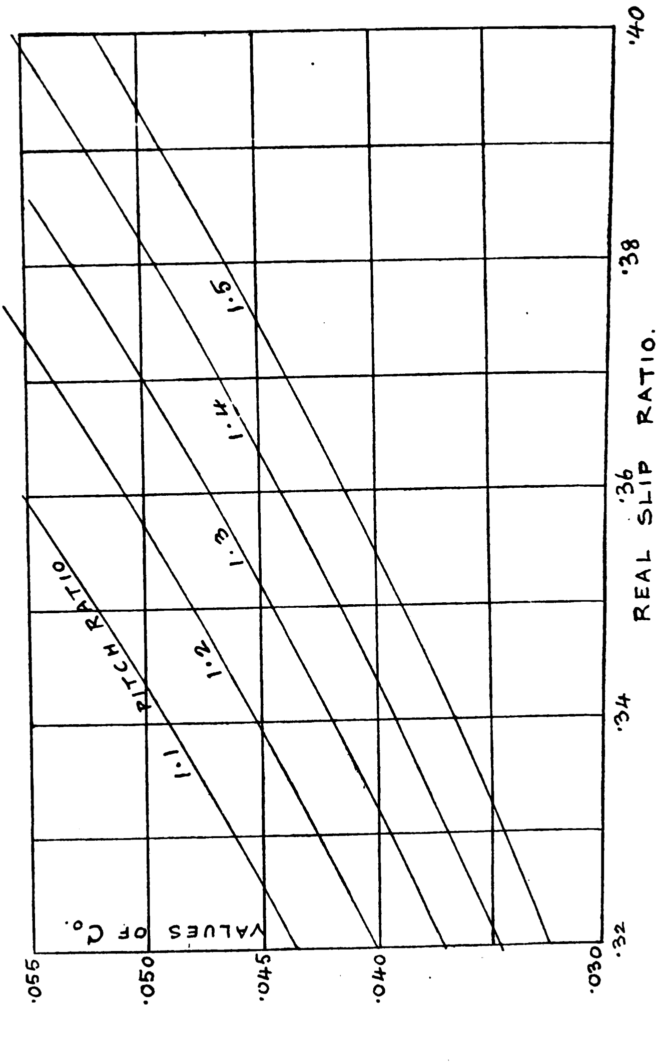


PLATE 62.



VALUES OF C_o.
REAL SLIP RATIO.
PITCH RATIO
1.1
1.2
1.3
1.4
1.5
.055
.050
.045
.040
.030
.32
.34
.36
.38
.40

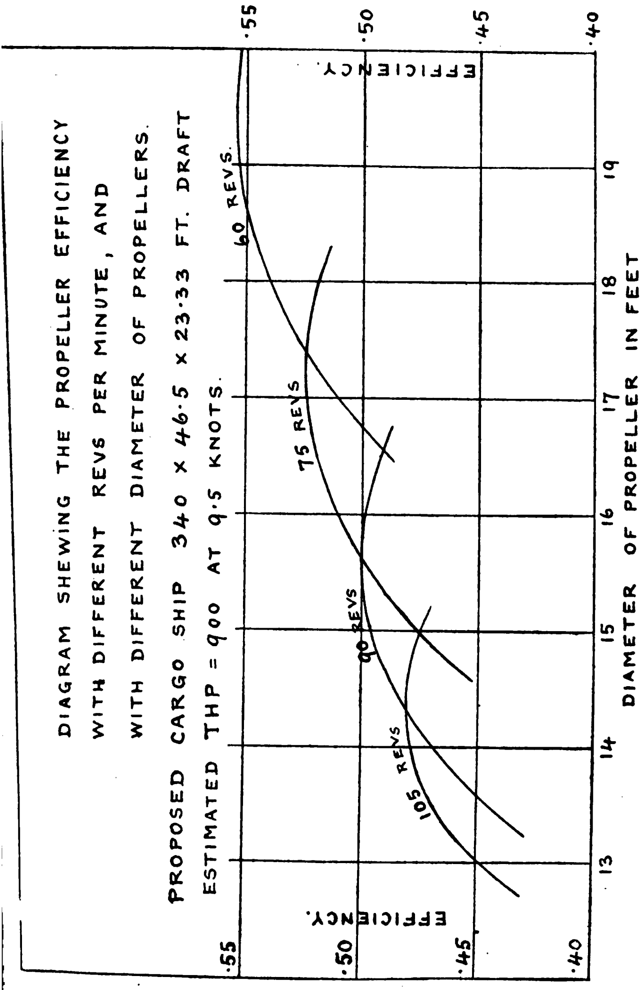
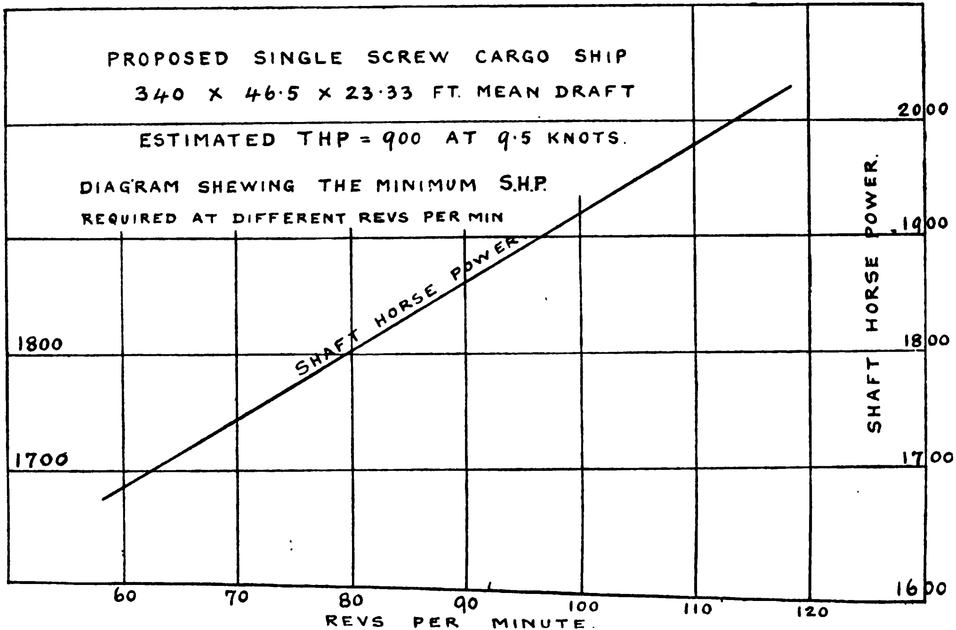
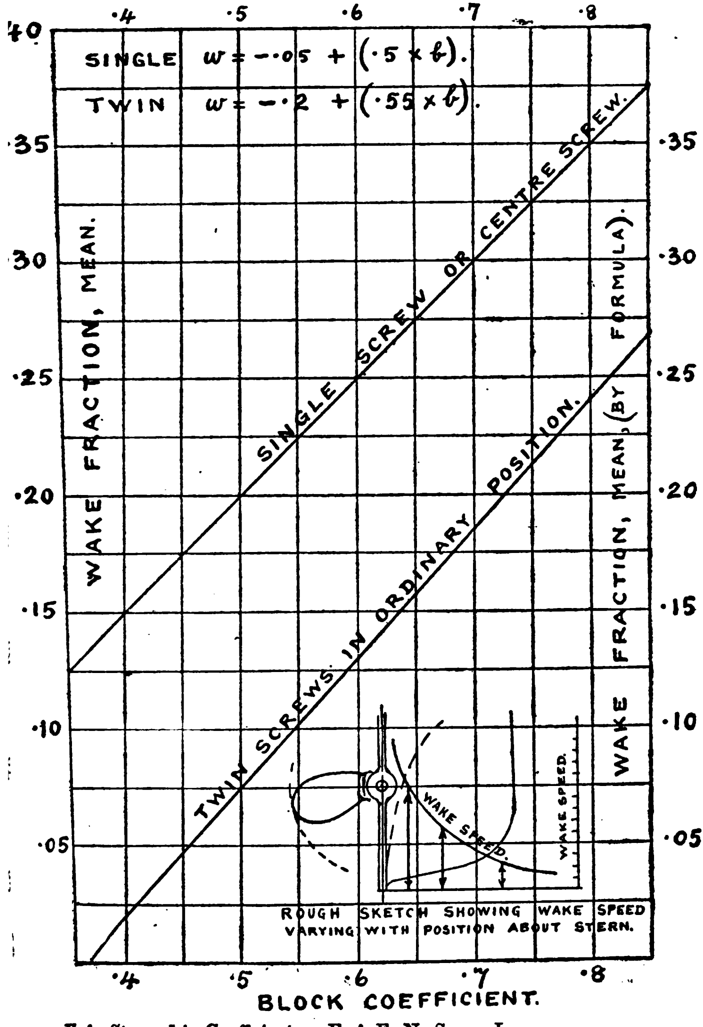
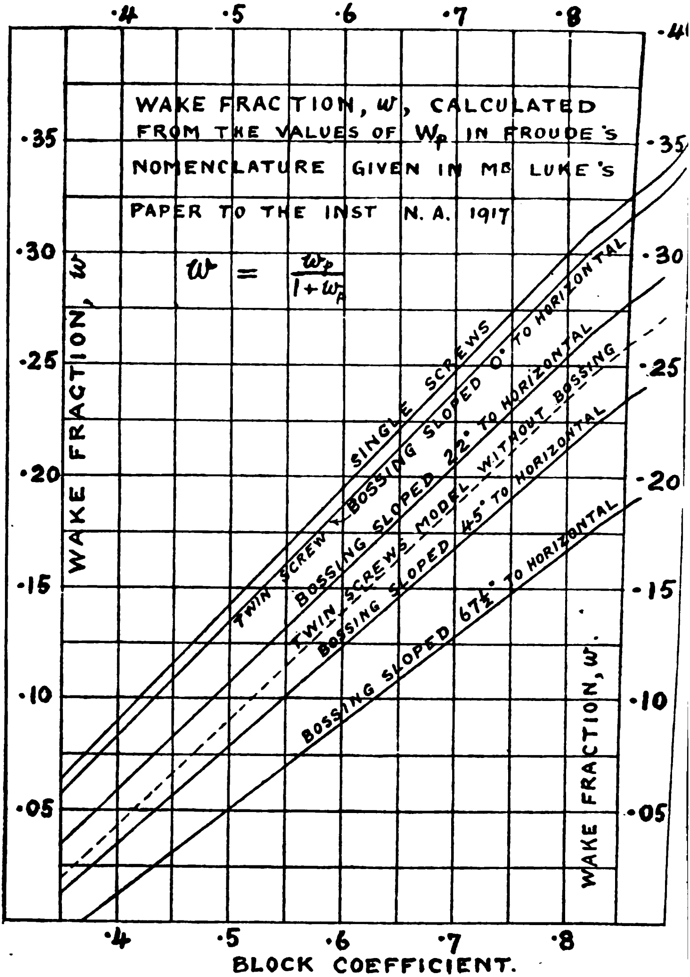


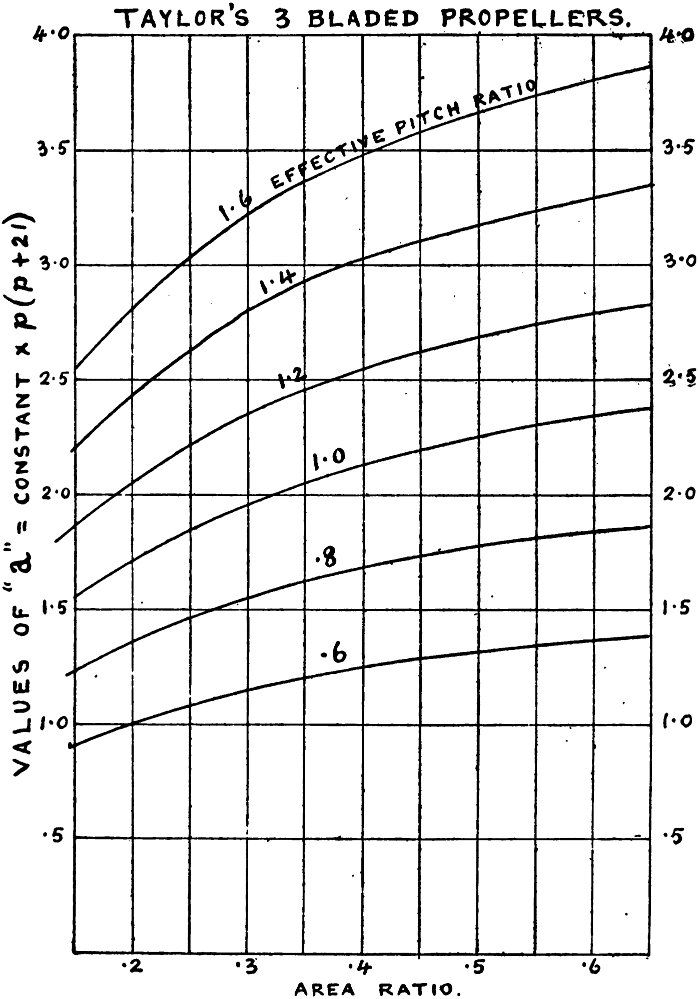
DIAGRAM SHEWING THE PROPELLER EFFICIENCY
WITH DIFFERENT REVS PER MINUTE, AND
WITH DIFFERENT DIAMETER OF PROPELLERS.
PROPOSED CARGO SHIP 340 x 46·5 x 23·33 FT. DRAFT
ESTIMATED THP = 900 AT 9·5 KNOTS.
EFFICIENCY.
·55
·50
·45
·40
60 REVS.
75 REVS
90 REVS
105 REVS
13
14
15
16
17
18
19
DIAMETER OF PROPELLER IN FEET


PROPOSED SINGLE SCREW CARGO SHIP
340 X 46·5 X 23·33 FT. MEAN DRAFT
ESTIMATED THP = 900 AT 9·5 KNOTS.
DIAGRAM SHEWING THE MINIMUM S.H.P.
REQUIRED AT DIFFERENT REVS PER MIN
SHAFT HORSE POWER
1800
1700
2000
1900
1800
1700
1600
SHAFT HORSE POWER.
60
70
80
90
100
110
120
REVS PER MINUTE.



Fyfe, Steamship Coefficients.—E. & F. N. SPON, LTD


WAKE FRACTION, w, CALCULATED FROM THE VALUES OF W_p IN FROUDE'S NOMENCLATURE GIVEN IN MR LUKE'S PAPER TO THE INST N. A. 1917
$w = \frac{w_p}{1+w_p}$
SINGLE SCREWS
TWIN SCREW + BOSSING SLOPED 0° TO HORIZONTAL
BOSSING SLOPED 22° TO HORIZONTAL
TWIN SCREWS MODEL WITHOUT BOSSING
BOSSING SLOPED 45° TO HORIZONTAL
BOSSING SLOPED 67½° TO HORIZONTAL
WAKE FRACTION, w
BLOCK COEFFICIENT.
·4
·5
·6
·7
·8
·35
·30
·25
·20
·15
·10
·05


TAYLOR'S 3 BLADED PROPELLERS.
VALUES OF "a" = CONSTANT × p(p+21)
AREA RATIO.
1·6 EFFECTIVE PITCH RATIO
1·4
1·2
1·0
·8
·6
4·0
3·5
3·0
2·5
2·0
1·5
1·0
·5
·2
·3
·4
·5
·6

MODIFICATION OF TOBIN'S DIAGRAM.